PRELIMINARY DESIGN OF A SELF-PROPELLED MANNED WORK PLATFORM

Constantine Pagonaris

NAVAL POSTGRADUATE SCHOOL
Monterey, California

THESIS

PRELIMINARY DESIGN OF A SELF-PROPELLED
MANNED WORK PLATFORM

by

Constantine Pagonaris

Thesis Advisor: E. A. McKinnon

September 1973

Approved for public release; distribution unlimited.

Preliminary Design of a Self-Propelled

Manned Work Platform

by

Constantine Pagonaris
Lieutenant, Greek Navy
Greek Naval Academy, 1963
B.S., Naval Postgraduate School, 1973

Submitted in partial fulfillment of the
requirements for the degree of

MASTER OF SCIENCE IN MECHANICAL ENGINEERING

from the

NAVAL POSTGRADUATE SCHOOL
September 1973

ABSTRACT

The maintenance operations of most large steel structures are currently being performed from permanent or movable platforms. The trend toward minimizing the cost resulted in a demand for a self-propelled movable platform that is controlled by the maintenance personnel on the platform. In this project a self-propelled work platform was created and designed which is capable of following the contour of a steel structure. The study was oriented toward naval applications where, in general, the ship hulls present curved surfaces which are often inclined with respect to the vertical. The proposed system can also be used or easily modified for non-naval applications.

TABLE OF CONTENTS

LIST OF FIGURES AND DRAWINGS

TABLE OF SYMBOLS AND ABBREVIATIONS

SYMBOLS

E	Modulus of Elasticity
G	Shear Modulus of Elasticity
I	Moment of Inertia
M	Bending Moment
T	Torsional Moment
t	Thickness
Z	Section Modulus
δ	Total Deflection
θ	Angle of Twist
μ	Sliding Coefficient of Friction
ν	Poisson's Ratio
σ_y	Yield Strength in Tension
σ_c	Compressive Strength
σ_s	Shear Strength
σ_a	Allowable Strength
σ_f	Flexural or Bending Strength
σ_t	Tensile Strength

ABBREVIATIONS

C.G.	Center of Gravity
Ksi	Kilopounds per square inch
lb, #	Pound Force
rpm	Revolutions per Minute
S.F.	Safety Factor

ACKNOWLEDGEMENT

The author wishes to express his sincere appreciation to Dr. Edwin A. McKinnon for the invaluable guidance, assistance and continuous encouragement which he provided during the pursuit of this study.

The author would also like to express his thanks to ERIEZ MAGNETICS who provided magnet data for this study.

Special acknowledgement must be given to my father, Zissis, who was my first teacher and who played an influential role in my career.

This thesis is dedicated to my wife, Vana, for her great help and understanding.

I. INTRODUCTION

Maintenance of steel structures requires access to the surface by personnel and equipment. The operations of cleaning, sand-blasting and painting are most efficiently performed from stable platforms. These platforms are generally of two basic types: permanent or temporary. A permanent platform such as a floor, a deck, or a catwalk provides an ideal working base. However, for most large steel structures such as ships and storage tanks available permanent platforms do not allow complete access to the working surface. This requires the use of temporary platforms which may be fixed or movable. The fixed platforms require the maintenance personnel to move their equipment as their work progresses and are generally expensive and time consuming to erect. The movable platforms do not present these problems and in general give the operators more flexibility in their choice of working area. This added flexibility becomes particularly important when only minor maintenance is to be performed at a number of locations on the structures surface. Movable work platforms often require the use of an auxiliary piece of equipment with its associated expense. Precise positioning of personnel and equipment can often be a difficult time-consuming operation if the structure has a complex geometry. This operation, of positioning personnel, can be simplified by

using a self-propelled movable platform that is controlled by the maintenance personnel on the platform.

In this project a self-propelled work platform was created and designed which is capable of following the contour of a steel structure. The study was oriented toward naval applications where, in general, the ship hulls present curved surfaces which are often inclined with respect to the vertical. The proposed system can also be used or easily modified for non-naval applications.

II. GENERAL BACKGROUND

In an effort to more completely define the problem and the areas for which the proposed system would have applications, it was necessary to review the techniques that are currently used by shipyards and the maintenance personnel of tank farms and water storage tanks. The results of this investigation have shown the existence of a variety of fixed and movable platforms of different degrees of sophistication. A better understanding can be obtained by presenting and discussing these methods separately.

A. SCAFFOLDING

Scaffolding is widely used for periodic maintenance and small repairs of steel structures, storage tanks, and smoke stacks. Figure 1 illustrates a custom designed scaffolding that is used to perform periodic maintenance on a power-plant stack. With the help of this rig a 22 foot diameter stack 165 feet high can be painted in 8 hours [1].* Scaffolding is also used in naval applications for cleaning and painting the surfaces of ships either in dry dock or afloat.

Cost is the main disadvantage of using scaffolds. Erecting scaffolds often require the help of several people other than those doing the maintenance work. Once in place,

---

*Numbers in brackets refer to list of references.

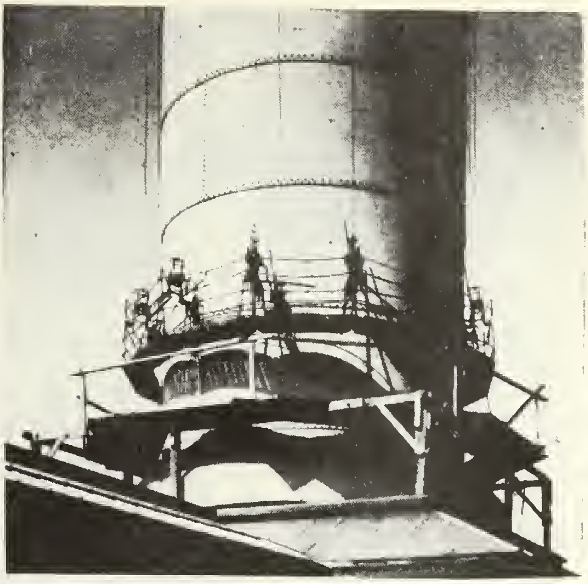


Fig. 1 Rigging for Stack Maintenance

generally only a small surface area can be reached depending, of course, upon the extent of the scaffolding.

B. STAGING

Staging is perhaps the most widely used method of providing access to the surface of large steel structures for periodic maintenance. In a modified form, staging is used for the cleaning and painting of large water storage tanks of fairly severe geometry as illustrated in Fig. 2 [2].

For the past half-century staging has been used in shipyards for the majority of the maintenance work on ship hulls with fairly efficient results. For major repairs or initial construction staging is still cost-effective. However, because of the time consumed in handling, erecting

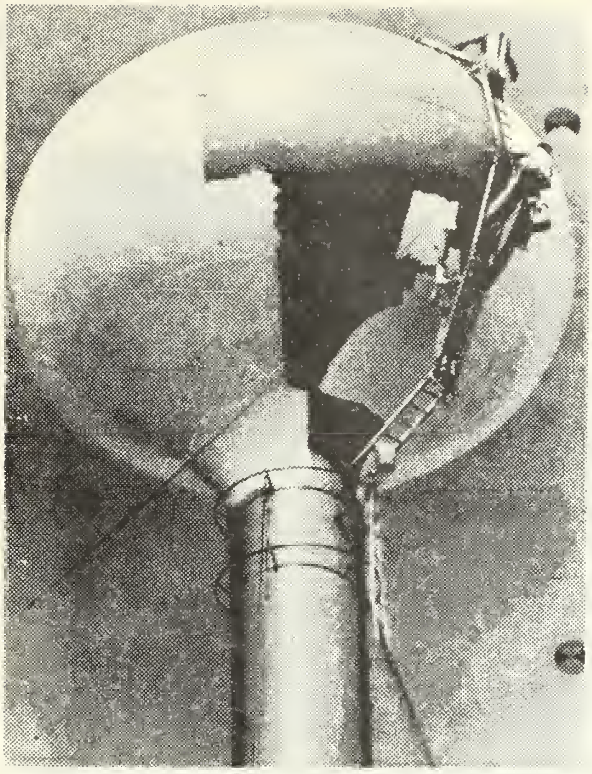


Fig. 2 Staging on a Water Tank

and disassembling staging and the high cost of labor, the use of staging for periodic maintenance is declining. Moreover, staging is sometimes very difficult, if not impossible, to use on steep surfaces as are often encountered on aircraft carriers. This difficulty is overcome by using scaffolding with the consequence of an even higher labor cost. Current interest in the U.S. Naval Shipyards is toward the use of self-propelled scissors-manlifts.

C. SCISSORS-MANLIFT

The scissor-manlifts are new units that have been recently introduced into most of the U.S. Naval Shipyards

[3]. They are self-propelled and hydraulically operated for vertical motion as illustrated in Fig. 3.

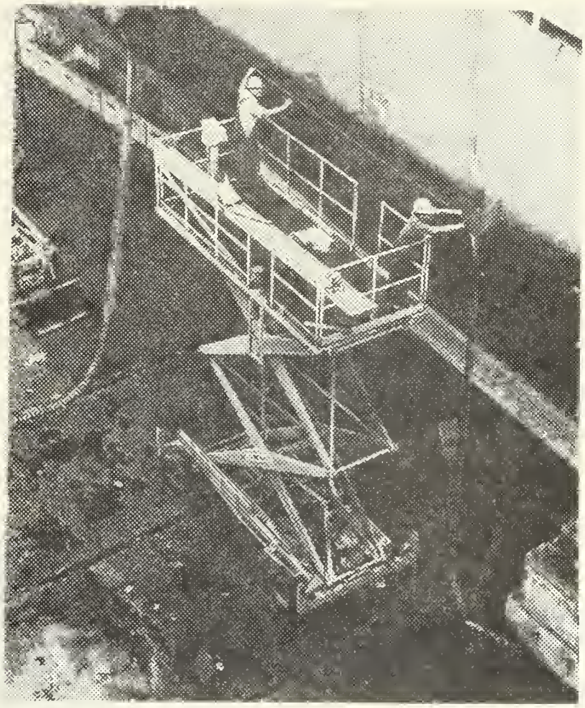


Fig. 3 Scissor-Manlift at a Dry Dock

The controls are located on the working platform and include the operating modes of forward, reverse, up, down and the actuation of the hydraulic outrigger stabilizers. One unit called a Scissors-Manlift comes in three models with a maximum working height of 26 feet, 31 feet or 40 feet and a drive speed range from 0 - 25 m.p.h. The chief limitations of these units are their working height and the need for a flat surface, free from obstacles, on which to maneuver.

D. PORTABLE SELF-PROPELLED SHOT BLASTING MACHINE

This machine was designed and engineered for the purpose of cleaning the shells of large storage tanks and ship hulls [2]. It travels horizontally or vertically over areas to be cleaned as shown in Fig. 4.

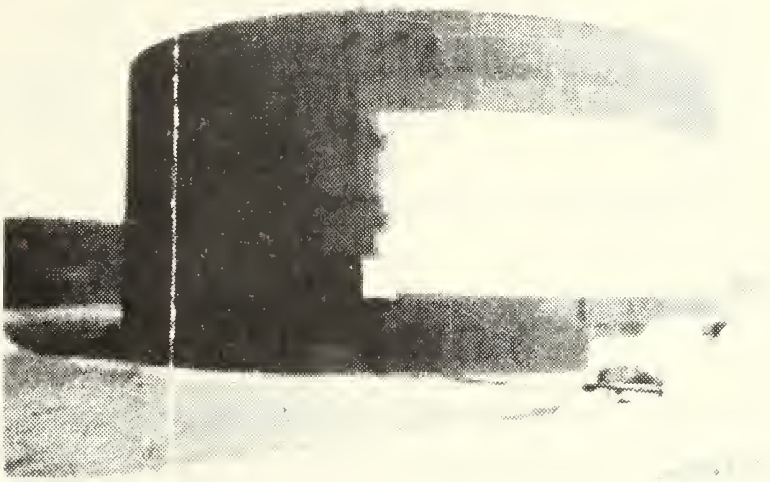


Fig. 4 Portable Shot Blasting Machine in Operation

This unit has a blast pattern of 36 inches in width and can obtain a good commercial blast at a rate of 40 square feet per minute. The machine can operate on ship hulls either in dry dock or when afloat and is operated by remote control thus eliminating the need for operators on the machine.

E. HULL INSPECTION PLATFORM

This unit has recently been constructed for underwater hull inspection [4]. The platform consists of a

self-propelled hull with a hydraulic crane attached as illustrated in Fig. 5.

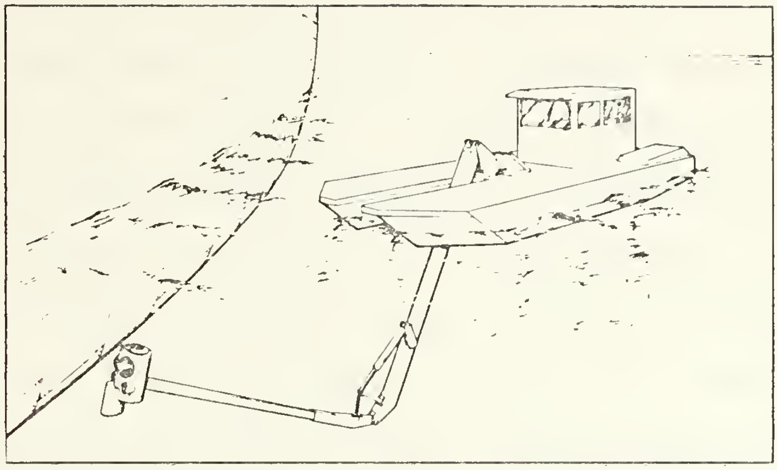


Fig. 5 Preliminary Design of Hull Inspection Platform

The capsule at the end of the crane supports an observer and it is lowered into the water providing a movable platform for the inspection of the ship hulls. The present model does not have any capability for hull maintenance operations.

III. DESIGN PROBLEM FOR NAVAL APPLICATIONS

For a design problem it is necessary to determine that an actual need does exist and to concentrate on the needs of the end user directly. A study was performed to find and analyze the problems that exist in the field of maintenance of steel structures. As this project is sponsored by the U.S. Navy, attention was focused on problems that currently exist in the Naval Shipyards [5]. With respect to the maintenance areas of sand-blasting and painting these problems can be summarized as follows:

a. Reduce the amount of labor required to clean and paint a hull.

b. Reclaim spent abrasives from shot blasting equipment and reduce the amount of dust produced to meet pollution standards.

c. Reduce the cost of placing operators close to steep surfaces on the ships hull, where staging becomes an ineffective technique.

d. Devise methods of applying uniform coatings of paint.

The cost of cleaning and painting a ship hull can be reduced by increasing the surface area an operator can handle in a day. This can be accomplished by providing better equipment for cleaning and painting and by decreasing the time required to move the operator and his equipment to a new area on the surface of the ship. A substantial

reduction of this time could be effected by the use of a self-propelled work platform that would follow the contour of a hull and be operated by the personnel on the platform, being supported from a dry dock wall or floor or from the side of the ship. A platform of this nature could be modified to accommodate a wide variety of equipment to perform the operations of welding, non-destructive testing, painting or sand-blasting with the particular aspect of preventing the contamination of the atmosphere. In addition, with special design the work platform could be used underwater for inspection, minor repairs or preliminary hull preparation prior to entering dry dock.

When the Naval Shipyards were confronted with this concept a very favorable response was received [3,6,7]. Such a device has not been developed to date by private industry because of the high development costs and the limited sales potential.

A. NEEDS

Before an actual design could be made a further analysis of needs had to be stated. This analysis has shown that the self-propelled work platform should satisfy as many of the following needs as possible:

a. Be self-propelled

b. Be controlled by the workmen on the platform

c. Be able to adhere to and maneuver on the surface of a ship

d. Be large enough to carry two men and their equipment

e. Incorporate the necessary safety precautions

f. Be adaptable for use in a dry dock

g. Be adaptable for use underwater

B. CONSTRAINTS

Certainly the self-propelled platform could not operate under all conditions. Therefore, the constraints of the system should be stated. These constraints were both operational and physical and can be summarized as follows:

a. Maximum load of 800 pounds including men and equipment.

b. Motion vertically from six (6) feet to sixty (60) feet above the dry dock floor.

c. Move vertically and horizontally at rates up to 30 feet per minute.

d. Operate both automatically and/or manually by a complete set of controls.

e. The device must be able to overcome small obstacles on the surface of the ship.

IV. DESIGN CONCEPTS AND SELECTION

The self-propelled platform can be broken down into four basic components; a bucket, a harness, a supporting structure and a power unit. In the design of these components several concepts and ideas were involved which constitute possible solutions. In order that the design needs would be satisfied, a feasibility study was performed to select the best design concepts for the final solution. For the purpose of clarity these components are discussed separately and the final solution presented at the end of this part.

A. BUCKET

The bucket provides the basic platform from which the maintenance personnel would work and be transported while also providing space for their equipment. It can be designed with special equipment for sand-blasting, or welding, or non-destructive testing or as a general all-purpose platform. For this design a general all-purpose platform was selected because it provided the greatest flexibility of use and the basis for future installation of automatic equipment.

B. HARNESS

The harness is attached to the bucket and holds it against the surface of the ship with an attractive force. This attractive force is necessary because often the surface

of a ship is inclined away from the vertical. Several concepts were considered for providing this attractive force and include the following:

1. Rotary Suction Cups

This concept uses the suction, produced by a vacuum pump to create an attractive force. This attractive force is transmitted to the surface of the ship by means of rectangular or oval shaped suction cups fitted on the circumference of a wheel. This concept has the advantage of being able to move on an inclined surface over relatively high obstacles, without interrupting the maintenance operation, being performed. However, an examination of a ships surface often reveals many small irregularities which means that the loss of vacuum is likely to occur.

2. Rotary Permanent Magnets

This device would consist of a wheel covered on the circumference with permanent magnets which would make contact with the steel hull and be capable of rolling on the surface while holding the bucket close to the surface. This device has the main advantage of being able to operate independent of the available power of the ship. However a preliminary design showed this concept to be unfeasible because of the unavailability of permanent magnets with a favorable weight-to-holding strength ratio [8].

3. Fixed Electromagnets

In contrast to permanent magnets, electromagnets have a favorable weight-to-holding power ratio. A rough design showed that it was possible to produce the required

attractive force while the weight of the device was kept within acceptable limits. Moreover, using the electromagnet under water would result in the same holding power as out of the water [9]. This concept was finally selected as the most promising one, although careful consideration must be given to any changes in the magnetic signature of the ship due to the induced magnetism of the electromagnets. A detailed examination of this problem will be discussed in Part V.

C. SUPPORTING STRUCTURE

The working platform and harness can either be supported from below as is the case of staging or of the Scissors-Manlift or suspended from above like scaffolding. Because of the maneuverability limitations of a system supported from below, it was decided to suspend the bucket and harness from a supporting platform which in turn would be attached to the ship. Several concepts were considered for attaching the supporting platform to the surface of the ship and will be reviewed below:

1. Free Moving Platform

For this concept the main deck of the ship was considered as the primary support. From the main deck a free moving platform could be used for the back and forth motion of the work platform. The up and down motion could be controlled by a system of pulleys and winches attached to the moving platform and the harness. This idea was

rejected because the maneuverability of the system is limited due to obstacles on the deck and in addition entire sections of the hull would be inaccessible because of superstructures. The problem of the obstacles could be overcome by using rail tracks on the main deck for the motion of the platform, thus improving the maneuverability since the motion would be independent of the obstacles. However the installation of these rail tracks, either welded on the surface or built-in on the main deck, presented the problems of high fabrication and maintenance cost and a safety hazard for personnel.

2. External Support

In this case the supporting structure could be temporarily or permanently installed on the free lateral surface of the ship, by any of the following means:

a. Welded on the hull

b. Intermittently welded on the lip of the main deck

c. Suspended from temporarily or permanently installed guides on the main deck

The use of an external support would give a maneuverability advantage to the work platform and the ability to approach most of the ship hull. However, it must be mentioned, that a permanent installation on some types of ships, although sometimes useful when the ship is afloat, is subject to damage if the ship gets too close to another vessel or structure. In this case a temporary installation is advisable.

This solution, satisfying most of the requirements, was selected for the supporting structure.

D. POWER UNIT

The power unit provides the means for the vertical and horizontal motion of the system. The selection of the individual components of this unit was dependent on the place of its installation, which in turn was closely associated to the final position of the supporting structure. Having fixed the latter, it was decided that a hanging platform from the external support would be a feasible location for the power unit. Presenting, here only the final solution, the power unit consists of:

1. Arrangement of motors, gear-boxes and driving wheels that would provide the horizontal motion.
2. A hoisting winch which via a system of pulleys would provide the vertical motion.

The available power of the ship or dry dock was used to run this unit. However it must be pointed out that this introduces a safety hazard in the event of a power loss. This problem could be overcome by using an autonomous unit for the power supply.

E. SUMMARY

Summarizing the final selection of each of the above components, the self-propelled work platform system can be described as:

a. An external support attached to the surface of the ship

b. A movable platform equipped with the power unit suspended form the external support

c. A magnetic harness, consisting of fixed electromagnets, which, by means of an arrangement of pulleys, will be suspended from the movable platform

d. A bucket which is the work platform connected to the magnetic harness

This final solution was selected because it satisfied most of the needs, and could be constructed from common structural materials using well established techniques.

V. DETAILS OF THE FINAL DESIGN

Attention will now be directed toward a detailed design of each part of the system. The basic criteria of this design was to keep the weight and cost at a minimum and to obtain the best performance and appearance. To accomplish these goals, the design variables, loading conditions and strength criteria of the complete unit were identified and then each member was designed separately. Compatibility checks were frequently made to prevent interference or mismatch between these parts. Environmental conditions were also considered for the material selection. The anticipated performance of the entire unit was evaluated in order to determine the number of needs that were satisfied. Design calculations of each component are shown in Appendix A. A general view of the unit is shown in Fig. 32.

A. BUCKET

One of the design needs was a manned work platform from which the operations of painting, sand-blasting, inspection etc. could be performed. The platform should be able to provide working room for men and equipment weighing up to 800 pounds and be able to move at fairly low speeds. Consistent with these needs, a decision was made to provide room for two workers, of an average weight of 200 pounds each, and 400 pounds in equipment to accomplish the desired maintenance functions.

These design goals were easily met by the bucket shown in Fig. 33. This bucket provides a working space for the two workers to perform their required operations and storage space for appropriate equipment. Room was left for the installation of control devices as remote operation was a requirement of the system.

A load analysis of the frame showed that tensile and compressive loads were significant while bending and torsion moment, although present, could safely be neglected. The design of the base and storage space was performed on a maximum allowable deflection basis, due to bending loads only. Since more than two loads were encountered the problem of finding the point of maximum deflection was complex. To simplify the problem a uniformly distributed load was considered and on the basis of that assumption the required moments of inertia were obtained, as shown in Fig. 6.

For this design weight and environmental conditions were the governing variables. The material selection was based on its strength-to-weight ratio and resistance to corrosion in a marine environment. Among the materials considered, wrought aluminum alloy 5086-H34 was selected because of its capability to be used in a marine environment without protection and its favorable strength-to-weight ratio. It is arc or resistance weldable by all commercial procedures and its workability is good [10].

Having completed the load analysis and material selection, the bucket can now be designed. The outside frame

consists of vertical square tubes interconnected by horizontal stiffeners to improve the stiffness and welded onto an aluminum base as shown in Figs. 35, 36. The base is composed of a series of channels covered by an aluminum sheet as shown in Fig. 37. The inside of the bucket was covered on the bottom two-thirds with sheet and the top one-third with expanded metal. The pivoting point of the magnetic harness was reinforced as shown in Fig. 41. The pivoting points for the magnetic harness and the ball screws were located by considering comfortable work positions for the operators. According to the available anthropometric data, a standing worker, of average height of six feet, can work comfortably if there is a distance of 1.5 feet from the working surface [11]. In Fig. 42 a graphical solution is shown with the extreme position of the working surface illustrated.

B. MAGNETIC HARNESS

The magnetic harness, the most vital component involved in this design, is composed of three main subcomponents; the structure, the electromagnets and the control mechanism. The harness provides the required attractive force by means of an arrangement of electromagnets to keep the work platform on the surface of a ship. The relationship between the attractive force, or holding power, and the gap, gap being the distance from the face of the magnet to the surface of the ferrous material, was examined in order to obtain the largest possible gap. The support structure was designed to hold the electromagnets and provide some degree of

flexibility because the contour of the ship surface must be closely followed. The required holding power of the magnets was kept at the minimum possible level, by means of a control mechanism, thus minimizing the effects of the induced magnetism.

1. Electromagnets

The selection of the electromagnets was based on the determination of forces required to keep the bucket on the ship surface. These equilibrium forces were obtained from a force analysis which involved three loading conditions, and several working positions from 0 to 40 degrees inclination. A free body diagram is shown in Fig. 10. The results of this analysis, shown in Tables 1, 2 and 3, indicated that the equilibrium forces behave differently for each loading condition and vary almost linearly as a function of the degree of inclination as illustrated in Figs. 11, 12, 13. The maximum force is of the order of 1,750 pounds at 40 degrees inclination at the upper part of the magnetic harness as shown in Fig. 13. Furthermore this analysis revealed that, for the bucket to stay in contact with the surface, an attractive force is always required for the upper part of this component whereas for the lower part, although this force is present, it is required under rather limited working and loading conditions, as shown in Fig. 12. Therefore an upper and lower set of magnets were used on the magnetic harness in order that a wider operational range be obtained.

The magnetic harness and the bucket had to be able to move on the ship hull. Wheels were used to hold the magnets away from the surface and provide the needed motion. The above analysis was again performed considering the equilibrium condition between the actual magnetic holding power and the reactions of the wheels and structure.

In this part of the analysis an assumption was made with respect to the actual distribution of the reactant forces on the structure and the wheels. By considering a linear variation of these forces the analysis was complicated because of a high degree of redundancy. It was, therefore assumed that taking an average condition would be acceptable for practical purposes, as shown in Fig. 14. Furthermore, it was obvious that the forces on the wheels were dependent on the applied holding power of the magnets. As this power had to be kept as low as possible, for reasons that will be discussed later, a minimum force of 25 pounds was assumed acting on each wheel. With these assumptions and combining all the loading and working conditions, the analysis, after the second iteration, showed that a holding power of 1,950 pounds was required for the upper set of magnets and 960 pounds was required for the lower set. Furthermore the operating envelope of the component and the maximum wheel reaction were determined. These results are plotted for the upper and lower magnetic sets as shown in Figs. 15, 16. The selection of the appropriate magnet could then be made as the required holding power was known. However, a brief

description of the actual characteristic of the magnet will facilitate this selection.

a. Performance of Magnet

The holding power of any magnet is a function of the distance from the face of the magnet to the surface of the ferrous material and the thickness of that surface. For the same thickness, the best performance is obtained when the magnet is completely in touch with the surface. The holding power almost exponentially decreases with an increasing gap, as shown in Fig. 17. Therefore the gap becomes a predominant factor for the design. It must be pointed out that in the gap, the thickness of the existing paint must be included which makes the situation even worse. Furthermore the thickness of the plate affects considerably the holding power as shown in Fig. 18, where the characteristic curves of two different types of magnets are plotted [12].

b. Selection of Electromagnets

The selection of electromagnets involves many problems that should be considered before making a decision. The magnetic signature of the ship should be kept at specified acceptable limits after the operation of the magnetic harness which in turn means that the magnetic forces should be as low as possible. A high gap was desirable in order that the system be able to overcome the small obstacles existing on the ship hull. The thickness of the working area varies from place to place and therefore

the holding power varies accordingly. Low weight and low cost in connection with the simplicity of construction and maintenance were also involved.

From the results of the force analysis a 2,910 pound magnetic force was required. This magnitude is associated with a rather low magnetic field as compared to that of the ship. However the induced field is rather randomly oriented which creates a problem [5]. Though expensive, excess induced magnetism can be removed by the degaussing system of the ship or by deperming stations. The variable thickness of the hull and the desired high gap were considered together with the required forces for the two sets of magnets in the harness. Figure 18 indicates that the Model 44 was designed for optimum holding with thinner materials and with maximum holding being obtained on 1/2 inch thick plate, whereas the Model 66 has a slightly lower holding power on thin sheets [12]. Since the hull is made of sheets up to 1/2 inch thickness Model 44 is the most promising. For the upper set four magnets were selected and two magnets were selected for the lower set. With six magnets a magnetic force of only 500 pounds each was required. From Fig. 17 this force is attainable either by using 1/4 inch plate and 1/8 inch gap or 3/16 inch plate and 1/16 inch gap. The latter combination was rejected because the gap was too small. Model 4418 was selected as satisfying all of the above requirements [12].

2. Support Structure

The structure of the magnetic harness served as a frame for the electromagnets and was connected to the bucket. One important condition imposed, was to provide some degree of flexibility to allow the magnets the ability to follow the contour of the ship. Therefore the failure criterion was based on strength only. Because many forces were applied to this frame, which vary because of changing load conditions, it was difficult to make a complete analysis of the whole structure. This analysis was done for each member of the structure making simplified assumptions as shown in Fig. 19. The acting loads were considered those obtained from the previous force analysis. Each set of forces shown in Tables 1, 2 and 3 was considered in order to obtain the worst working and loading conditions upon which the design would be based. Loading condition No. 3 at an inclination position of 40 degrees was found to be the worst. A combination between strength and deflection could be obtained by selecting the proper material. However, since the magnets were made of steel the same material was selected for the structure to reduce galvanic corrosion problems. The required flexibility was achieved by selecting the appropriate cross-section of each member and incorporating a flexible spring system.

During selection of the structural members, efforts were made to minimize the weight. These members are joined together by welding to achieve high strength and negligible

weight addition. Because low stresses were encountered, low-carbon structural steel was selected. The final design is shown in Figs. 43, 44, 45.

The point of connection of the magnetic harness with the bucket presented a corrosion problem because aluminum and steel were in contact. This problem was overcome with a Teflon sleeve and washers which separated the steel and aluminum surfaces. To complete the unit, four pulleys were installed on the horizontal arm, as shown in Fig. 33. A cable running from the moving platform down through these pulleys and then back up to a winch on the moving platform was used to control the vertical motion of the bucket and harness as illustrated in Fig. 32. It should be pointed out that this design, in connection with the pivoting point of the bucket, provides comfortable working positions for the workers up to an inclination of 40 degrees, as shown in Fig. 42.

3. <u>Selection of Wheels</u>

From the previous analysis and Fig. 15 the maximum load on each wheel was 200 pounds. This load was obtained for an extreme working condition of 40 degrees and on the assumption of 1/4 inch working plate and 1/8 inch gap. If the thickness of the plate were to increase for the same gap, the force on each wheel would also increase. However, the 200 pound force represented an extreme situation and taking into consideration the imposed safety factor of the manufacturer, this maximum load was considered acceptable

for the selection of the wheels. A swivel type caster with hard rubber tires was needed to allow vertical and horizontal motion and to maintain the desired air gap. Moreover for the purpose of appearance, a rather low caster was desired to match the design. The BASSICK SWIVEL CASTER NO. 3771 was selected satisfying the above requirements [13].

4. Control Mechanism

The force analysis has shown that the required holding power of the magnet is a function of the angle of the surface inclination. This function would be useful in determining the required holding power for each working position and could be obtained by means of a control mechanism. For the purpose of evaluating the forces on the magnetic harness a linear function was assumed as shown in Figs. 15, 16. However, for the purpose of designing a control mechanism the exact function must be determined which could be obtained using well known numerical techniques. Since this was not one of the objectives of this present study, no exact functions were determined. However, several ideas will be presented which are capable of controlling the magnetic power.

a. One concept consists of a device which measures the angle of inclination of the magnetic harness and a control device which, upon receiving this measurement, would regulate the current to the magnets to obtain the required holding power. This angle could be measured between the bucket and the magnetic harness since the former always

remains horizontal and the harness has to follow the contour of the ship. The controlling device would operate on a transfer function based on the functional relationship between power and angle.

This control mechanism, could not be used when the self-propelled unit is used on a ship at sea because the angle measurement is relative and requires the ship to stay fixed.

b. Another method is based on the relationship between the magnetic power, the wheel reaction and the angle of inclination, as shown in Figs. 15, 16. Load cells could be installed on the caster supports capable of measuring the acting forces during operation. The magnitude of these forces would be received by the control device. To operate this mechanism a preset value would be introduced in the control device for each load cell and the current to the magnets would be regulated according to the relationship between actual and desired wheel reactions. This mechanism presents the disadvantage of being fairly complex and subject to many small perturbations in the loads as the operators shift their weight in the bucket and as the harness moves over a rough surface.

5. Ball Screw

In addition to the pivoting point, shown in Fig. 41 connecting the bucket and the magnetic harness, it was necessary to provide a rigid connection between them, to ensure stability of the work platform. In addition, this rigid

connection was desirable for two reasons, first, to avoid the crushing impact of the magnetic harness on the bucket in the case of power loss, and second, to obtain better distribution of stresses in these two components. These requirements were accomplished by means of two ball screws, one on each side of the bucket, as shown in Fig. 33. The force analysis revealed that a maximum force of 430 pounds would be experienced by each ball screw. Also from Fig. 42 the extension of each ball screw would be in the range of 18 to 25 inches. Duff-Norton Mini-Pac mechanical actuators of 500 pound capacity were selected which satisfied the above requirements and in addition are very compact and light, easily installed, and provided with a brake mechanism.

The motion of the ball screws would be synchronized with the motion of the magnetic harness to allow the bucket to remain horizontal. This could be obtained by electrically connecting the ball screw with the magnetic harness control mechanism.

C. MOVING PLATFORM

The moving platform provided a base for the horizontal and vertical drive units and a support for the bucket and magnetic harness. The moving platform in turn was suspended from the upper moving supports as shown in Fig. 32. The moving platform itself was restricted to move on only nearly vertical portions of the ship hull. An iterative procedure was followed for the design of the platform base and the selection of the appropriate components of the drive units.

1. Base Design

A preliminary force analysis revealed a complex combination of bending moments and vertical and horizontal forces. The design of the base consists of two main steps; first the main structural members were designed to resist the applied loads, and second the top panel was designed to resist localized bending between the main structural members. Because of the complexities of the bending and a desire for a stiff platform base, a maximum unit deflection of 0.0001 inch per inch was chosen as a design criterion. For the purpose of simplicity, several assumptions were made with respect to the application of forces as shown in Fig. 24.

The cross-section, shown in Fig. 24(b), was found to have sufficient moment of inertia to resist the vertical bending load without excessive deflection. Lateral loads and localized bending were also considered as shown in Figs. 25, 26 respectively.

For this design, rigidity and weight were the predominant variables. Among the commercially available materials mild steel ASTM-A36 was selected because it is the lightest, most economical metal for equivalent rigidity on a weight-to-weight basis. It is weldable by all commercial procedures and it has good workability.

The platform base is shown in Figs. 47, 48, 49. It consists of three longitudinal channels welded to two transverse channels at the ends. On top of this frame a 1/8 inch

diamond plate was welded. To improve stiffness, two transverse rectangular bars were added. Two metallic "boxes" were provided for the driving wheels as shown in Fig. 49. Four pulleys were added for the cable of the magnetic harness and four supports were provided to suspend the platform from the moving supports.

2. Drive Units

The drive units consist of two main sub-components; the driving unit associated with the horizontal motion and the lifting unit providing the vertical motion.

a. Driving Unit

Having designed the base, all external forces were fixed, as shown in Fig. 22. From an analysis the supports and surface reactions were obtained. The friction or driving force could then be obtained provided that the rolling coefficient of friction was known. Examining the ship hull it can be observed that the condition of the surface depends not only on the working location but also on the atmospheric conditions. This subsequent variation in the coefficient of friction changes the available driving force, if the wheels slip. For the purpose of design, a sliding coefficient of friction of 0.05 was assumed, which corresponds to rubber on wet steel [14]. Using this value for the coefficient of friction and a horizontal speed of 30 feet per minute a power requirement of 0.21 hp was obtained.

For the above power and speed requirements many possible arrangements of commercially available speed reducers and motors exist. The configuration shown in Fig. 47 was chosen because it was compact, economical, easily installed and maintained. It consists of the following items: [13, 15, 16, 17].

1. Two speed reducers, Morse 18GCDV,
   Input: rpm 1750
   Output: rpm 14.0, Torque 561 in-lbs.
2. Two electric motors, 1/4 h.p., Morse 25TE115
3. Remote control Morse MA-25
4. Two driving wheels Bassick No. WR-1248
5. Two rigid couplings Dodge R16
6. Four roller bearings SKF 478204-012

b. Lifting Unit

The lifting unit used to raise and lower the bucket and magnetic harness is a winch capable of providing maximum vertical motion of 30 feet per minute. From the force analysis, shown in Fig. 10 a maximum force of 750 pounds was required to support the bucket. An automatic brake was desirable for safety requirements and because the bucket was suspended by two electric hoist cables a divided drum was necessary. Beebe No. 800 A20 was selected for this purpose equipped with remote controls [18].

It should be noted that the remote controls of both the driving unit and the hoist will be installed in the bucket.

D. MOVING SUPPORTS

The moving support is the component from which the entire unit is suspended, as shown in Fig. 32. Two identical and independent supports were provided for this purpose. Each of these supports consists of a frame, wheels and hanger. A simplified stress analysis revealed that 1/8 inch steel plate AISI M1020 was satisfactory. Four main wheels were used for each support, two were used to carry the transmitted vertical loads and provide stability for the support. Two more were eccentrically mounted to resist the horizontal force and bending moments applied to the support from the moving platform. In addition, four small guide rollers were used to direct the support along the guideway mounted on the surface of the ship. A general view of a moving support is shown in Fig. 51. A hanger, attached to the top of each support, was the method used to connect the support with the moving platform and is shown in Fig. 55. This hanger provides two pin joints for the supporting bars of the moving platform and thus connects the support with the platform.

E. EXTERNAL SUPPORT

The external support is the rail track for the moving support and is welded on the ship hull, as shown in Fig. 32. Loads are transmitted to the external support by the vertical and horizontal wheels of one of the moving supports as shown in Fig. 30. A simplified analysis of this loading condition was performed as illustrated in Fig. 31. Steel

plate C-1045 of 1/2 inch thickness was found satisfactory. For the present study this support was considered as a permanent installation. However, it should be pointed out that this type of installation, although sometimes useful when the ship is afloat, is subject to damage when the ship gets too close to another ship or structure. Depending on the type of ship, the external support can also be installed temporarily by being intermittently welded on the lip of the main deck or suspended from guides on the main deck.

F. GENERAL COMMENTS ON DESIGN

In the above analysis loads and environmental effects were not considered. Therefore, it was decided to use higher factors of safety than necessary for static loading to account for dynamic loading and corrosion. The type and size of welds were not included in this design, although there would be no problem as the materials selected are weldable by most procedures. Moreover the allowable stresses were based on the effect of welding. The total weight of the system including personnel and maintenance equipment is 2,000 pounds.

One important safety feature needs to be examined closer. If a power loss occurs, the unit will be held by cables on the magnetic harness and the automatic brake of the hoisting winch. However, two potential problems exist: first, the unit may swing away from the surface of the ship and might hit another piece of equipment or a dry dock wall, and second,

an overturning is likely to occur. Although these two problems were not examined in detail, recommendations will be stated which give possible solutions.

The first problem can be overcome by one or a combination of the following two solutions:

a. An emergency power supply system consisting of series of batteries and a protection relay can be connected to the main power plan. In the case of power loss in the main plan the relay will actuate the batteries to provide the required current for the operation of the magnets. This idea is made feasible by the existence of residual magnetism in the electromagnets which will hold the harness against the ship until the auxiliary power unit takes over.

b. A second solution consists of a winch capable of moving on a rail track installed on the floor of the dry dock. This winch would be equipped with a brake mechanism and provide the required cable which would be connected to the bucket. The electric system of the winch would be connected to that of the unit and the winch motion would be automatically controlled by control devices in the bucket. The rail track would be installed close to the lowest portion of the hull. During normal operation the cable will be in slight tension. In the case of power loss the brake mechanism of the winch would lock the cable which in turn will exert an inward force on the bucket thus preventing the unit from swinging.

The solution of the problem of overturning could consist of a solenoid device installed on the four pulleys of the magnetic harness. During a loss of power, the device would lock the cable running through these pulleys thus preserving the required stability and avoiding the overturning.

VI. CONCLUSIONS AND RECOMMENDATIONS FOR FUTURE WORK

The unit developed and designed in this project is a self-propelled manned work platform adaptable for naval applications while a modification is feasible for water tanks, metallic smoke stacks, etc. The operation of the unit does not require other supporting personnel or equipment. An autonomous unit for the power supply, installed on the moving platform, could make the system completely independent of the ship and dry dock. Space is provided for two workers and appropriate equipment to perform the operations of inspection, sand-blasting, painting or cleaning of a ships hull. The unit is operated automatically and/or manually by a complete set of controls, located in the bucket, for the horizontal and vertical motion. Speeds up to 30 feet per minute in both directions allow access to a fairly large surface area per day. The work platform is capable of adhering to and maneuvering on inclined surfaces of a ship up to 40 degrees by means of the magnetic harness. There are no limitations on the height of the working surface provided the hoisting winch is furnished with the required cable. Since water does not influence the performance of the magnets, the unit is capable of operating underwater provided appropriate insulation precautions are taken.

The structure of the self-propelled platform consists of standard structural shapes of aluminum and steel. The

fabrication can be performed by well-established techniques.

The unit can be modified to accommodate a wide variety of equipment. As the platform follows the contour of the ship a system of nozzles, at an angle with respect to the hull, could be installed on the magnetic harness frame for sand-blasting operation. A vacuum could be generated around this system of nozzles for dry recovery thus reducing the operation cost and the amount of dust. A similar procedure could be followed for painting. Furthermore, if more than two workers are desirable during an operation, the installation of a simple support between two units is also feasible.

Recognizing that this project was a preliminary design several interesting projects for further improvement of the unit will be recommended.

a. The unknown influence of the operation of the magnets on the magnetic signature of the ship was a potential problem that was considered. The reason for this was that the imposed magnetic field was randomly oriented and of unknown intensity. An experimental study or simulation of the system could determine the pattern and intensity of this field and if warranted, more elaborate means to face the problem can be considered. If the intensity of this field can be safely increased, a higher attractive magnetic force can be allowed and consequently the gap can be increased, thus permitting the system to operate over higher obstacles on

the ship hull. This study would constitute an interesting project for further improvement of the unit.

b. The detailed design of the electric system of the unit and the appropriate control mechanisms have to be done. It should be pointed out that a fast response of the control mechanism associated with the angle of inclination of the magnetic harness is highly desirable. In this project consideration should also be given for underwater operation of the system.

c. As this unit provides an improvement of the present operation in dry docks, the building of a model would constitute an interesting project for the evaluation of the actual performance of the platform. This would allow experimental testing and indicate desirable modifications that would improve the performance of the unit and increase the areas of applications. For this project a complete set of mechanical and electrical drawings is desirable.

APPENDIX A - DESIGN CALCULATIONS

A. BUCKET CALCULATIONS

Material: Aluminum Alloy 5086-H34 with mechanical properties: [19]

$\sigma_y = 28$ ksi, $\sigma_s = 16$ ksi, $\sigma_a = 12.5$ ksi, $E = 10{,}400$ ksi

1. Base of Bucket

a) A model of the bucket base is shown in Fig. 6(a). Consider portion ABCD and assume a plate of thickness $t = 0.125$ in. with three edges simply supported and one fixed, with a uniformly distributed load over the entire surface $w = 0.231$ lbs/in^2, as shown in Fig. 6(b).

The maximum stress and deflection are: [20]

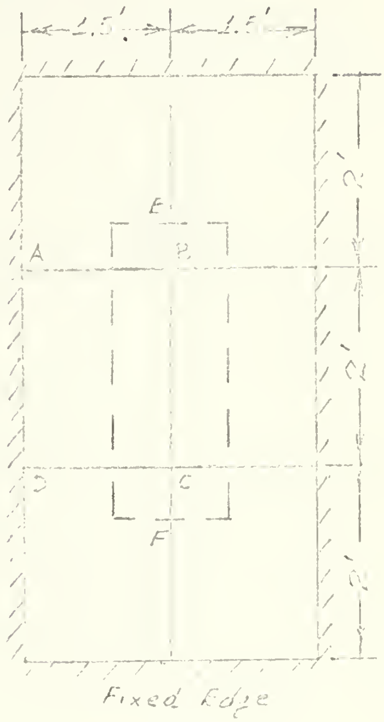
$$\sigma_{f_{max}} = \beta \frac{wb^2}{t^2} = 2630 \text{ psi}$$

$$\delta_{max} = \alpha \frac{wb^2}{Et^3} = 0.004 \text{ in}$$

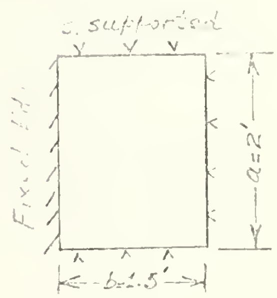
with allowable stress $\sigma_a = 12.5$ ksi and deflection $\delta_a = 0.1$ in., the factors of safety are:

$(S.F)_f = 4.75$ and $(S.F)_\delta = 7.5$

∴ Use 1/8 in. diamond tread aluminum plate.



(a)



(b)

Fig. 6 Base of Bucket

b) To evaluate the longitudinal and transverse supports of the bucket base shown in Fig. 6(a) consider a portion of the longitudinal support as a built-in beam at both ends with a uniform load $w = 4.2$ lbs/in. as shown in Fig. 7(a). The cross section, shown in Fig. 7(b), consists of a channel welded on the 1/8 in. base plate. Total moment of inertia $I = 0.02 \text{ in}^4$.

The maximum stress and deflection are:

$$\sigma_{f_{max}} = \frac{w\ell^2}{12Z} = 1340 \text{ psi}$$

$$\delta_{max} = \frac{w\ell^4}{384EI} = 0.09 \text{ in}$$

The factors of safety for the same allowable stress and deflection are:

$(S.F)_f = 9.3$ and $(S.F)_\delta = 1.12$

∴ Use al. ⊔ 1" x 1/2" x 1/8"

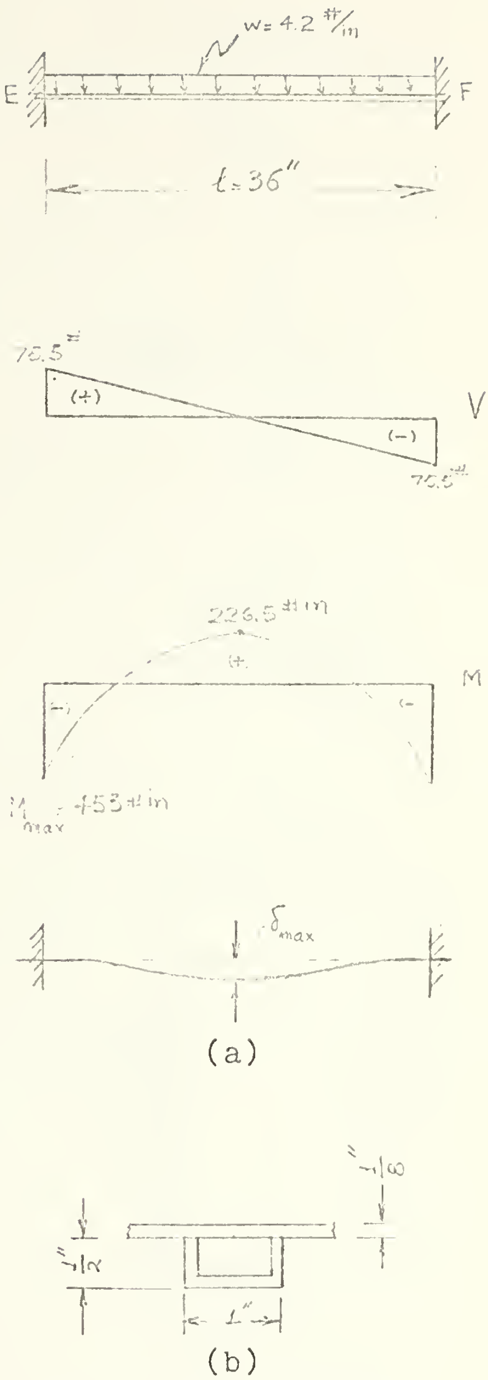


Fig. 7 Longitudinal Base Support

2. Storage Space

a) An outline of the storage space is shown in Fig. 8(a). Consider portion (I) as a plate of thickness 1/8 in. with a uniformly distributed load w = 0.14 lbs/in^2 over the entire surface and constrained as shown in Fig. 8(b).

The maximum bending moment, stress and deflection are given as: [20]

$$M_{max} = \beta w b^2 = 38.6 \text{ lb-ins at point C}$$

$$\sigma_{f_{max}} = \frac{\beta w b^2}{t^2} = 2450 \text{ psi}$$

$$\delta_{max} = \alpha \frac{w b^4}{E t^3} = 0.092 \text{ in. at point D}$$

The allowable stress and deflection are the same as for the bucket. The factors of safety are:

$(S.F)_f$ = 5 and $(S.F)_\delta$ = 1.1

∴ Use 1/8 in. aluminum plate

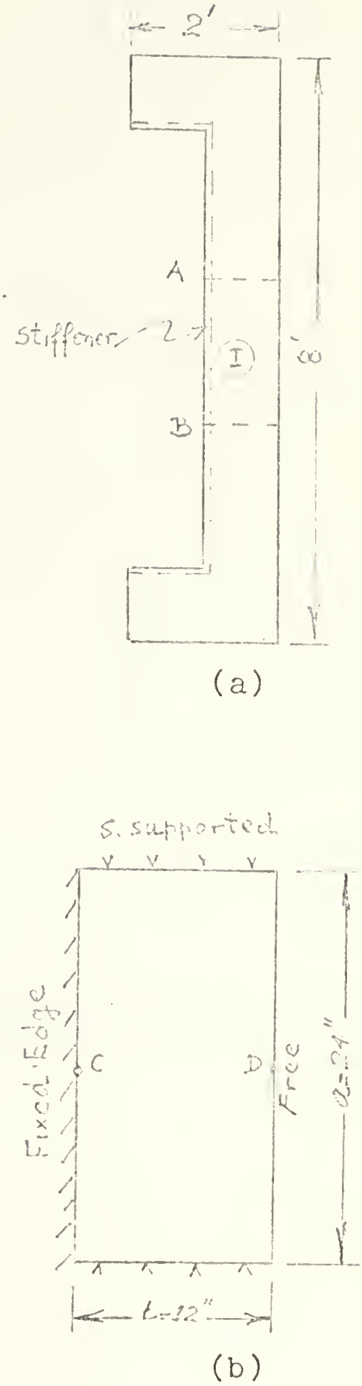


Fig. 8 Storage Space

b) At the intersection of the storage space with the bucket a horizontal stiffener was provided.

Consider portion AB, shown in Fig. 8(a), as a built-in beam acted on by a concentrated torsional moment as shown in Fig. 9(a). Using rectangular tubing 1" x 1" x 1/8" shown in Fig. 9(b) the maximum shear stress and angle of twist are given by: [20]

$$\sigma_{max} = \frac{T}{4t(a-t)^2} = 100 \text{ psi}$$

$$\theta_{max} = \frac{T\ell}{2KG} = 0.002 \text{ rad.}$$

where: t = thickness of tubing

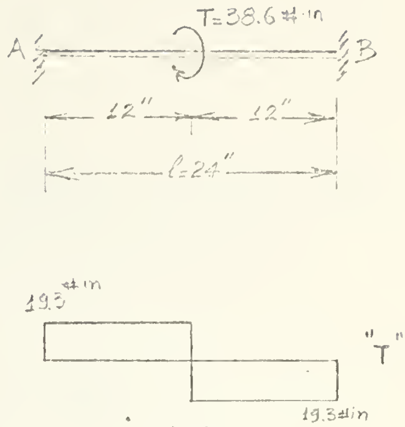
$K = t(a-t)^3$

$G = \frac{E}{2(1+\nu)}$

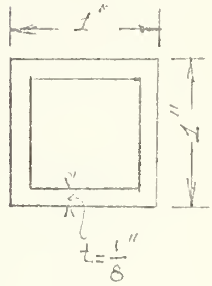
$\nu = 0.316$

∴ Use square aluminum tube:

1" x 1" x 1/8"



(a)



(b)

Fig. 9 Horizontal Stiffener

B. MAGNETIC HARNESS CALCULATIONS

1. Force Analysis

a. Equilibrium Forces

One configuration of the magnetic harness and bucket is shown in Fig. 10, at an arbitrary inclined working position of angle ϕ.

The following notation was used:

F_c = cable reactive force

N_1 = upper equilibrium force

N_2 = lower equilibrium force

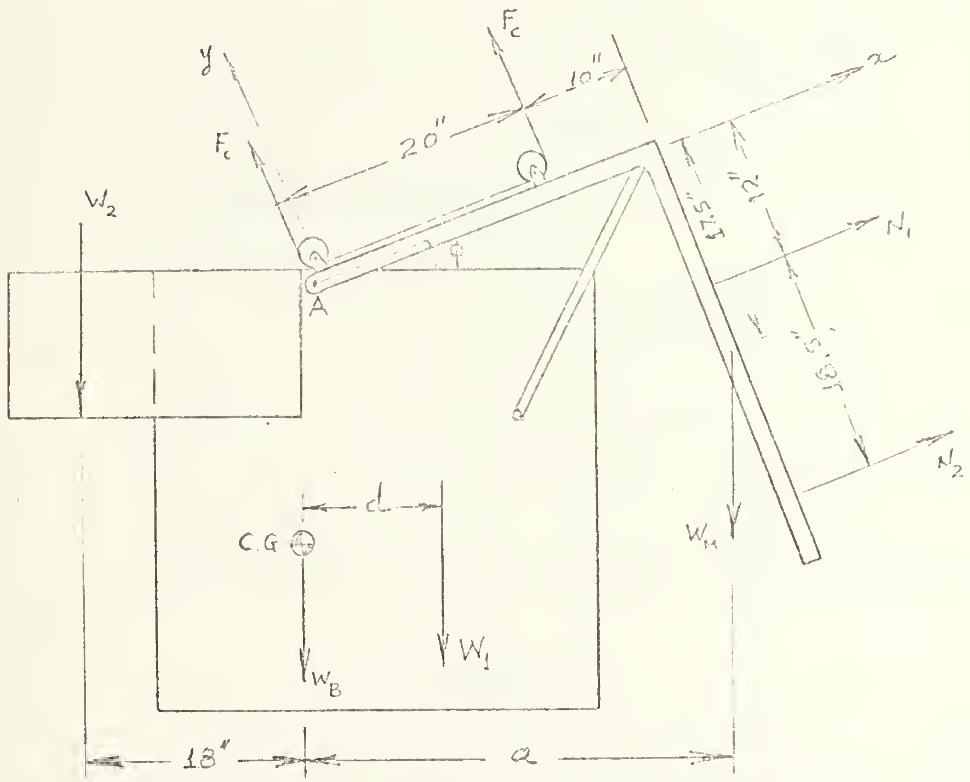


Fig. 10 Free Body Diagram of Bucket and Magnetic Harness

W_B = bucket weight (200 lbs)

W_1 = weight of men and equipment

W_2 = weight of equipment

W_M = weight of magnetic system (500 lbs)

a = distance of center of gravity of magnetic system to center of gravity of bucket.

Three loading conditions were examined, as follows:

No. 1 : W_1 = 600 lbs, W_2 = 200 lbs, d = 6 inches

No. 2 : W_1 = 600 lbs, W_2 = 0 , d = 20 inches

No. 3 : W_1 = 600 lbs, W_2 = 200 lbs, d = -8 inches

The variable distance a is given by:

$$a = 30 \cos\phi + 17.4 \sin\phi \,.$$

(1) <u>Loading Condition No. 1</u>

Equilibrium equations:

$$\Sigma F_x: \quad N_1 + N_2 = 1{,}500 \sin\phi$$

$$\Sigma F_y: \quad F_c = 750 \cos\phi$$

$$\Sigma M_A: \quad N_1 + 2.54 N_2 = 41.6a - 1.67\, F_c$$

Results are shown in Table 1 and plotted in Fig. 11.

TABLE 1

ϕ	a	N_1	N_2	F_c
0	30.	0	0	750
15	33.5	520	-132	724
30	34.7	1004	-254	650
40	34.2	1288	-324	574

(2) Loading Condition No. 2

Equilibrium equations:

$$N_1 + N_2 = 1{,}300 \sin\phi$$

$$F_c = 650 \cos\phi$$

$$N_1 + 2.54N_2 = 41.6\ a - 1.67\ F_c + 1{,}000$$

Results are shown in Table 2 and plotted in Fig. 12.

TABLE 2

ϕ	N_1	N_2	F_c
0	-764	764	650
15	-322	656	628
30	95	555	563
40	343	492	498

(3) Loading Condition No. 3

Equilibrium equations:

$$N_1 + N_2 = 1{,}500 \sin\phi$$

$$F_c = 750 \cos\phi$$

$$N_1 + 2.54N_2 = 41.6\ a - 700 - 1.67\ F_c$$

Results are shown in Table 3 and plotted in Fig. 13.

TABLE 3

ϕ	N_1	N_2	F_c
0	454	-454	750
15	974	-586	725
30	1458	-708	650
40	1745	-777	575

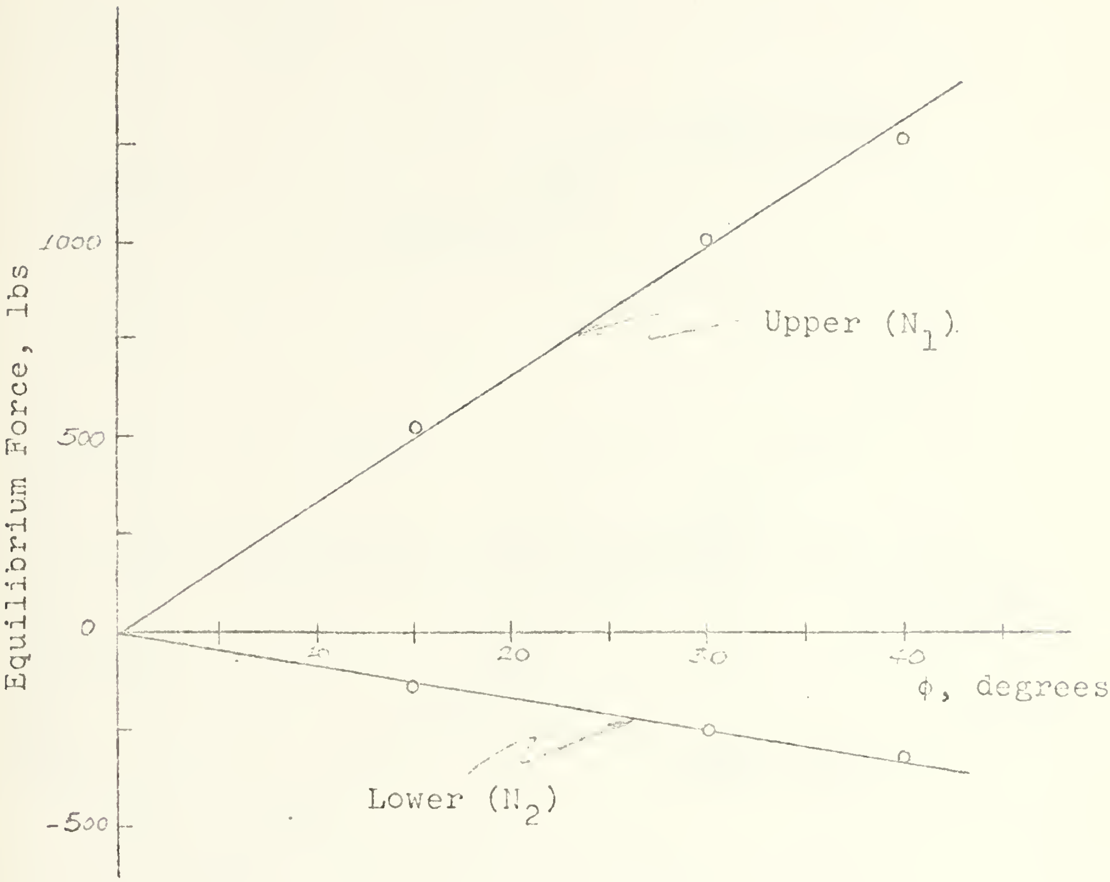


Fig. 11 Equilibrium Forces vs Surface Inclination — Loading Condition No. 1

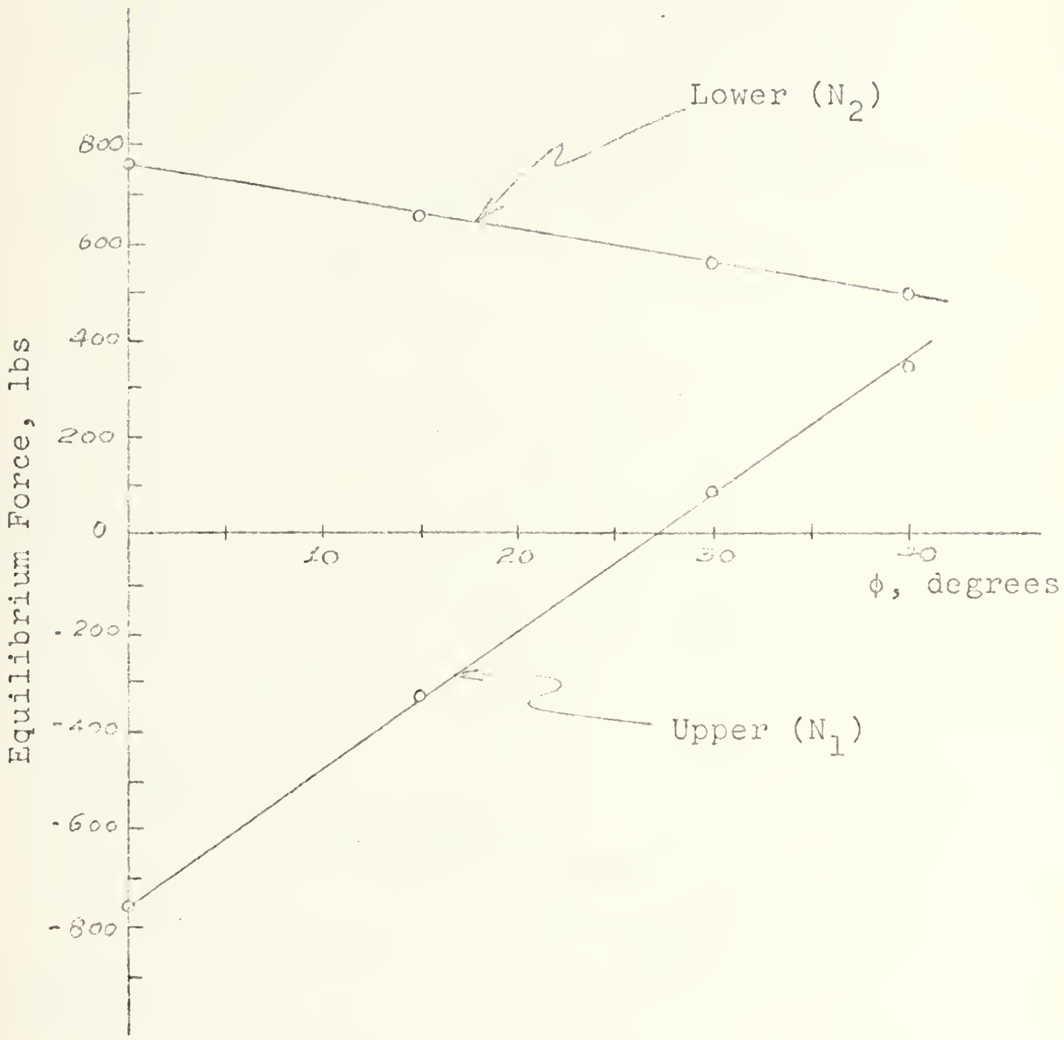


Fig. 12 Equilibrium Force vs. Surface Inclination – Loading Condition No. 2

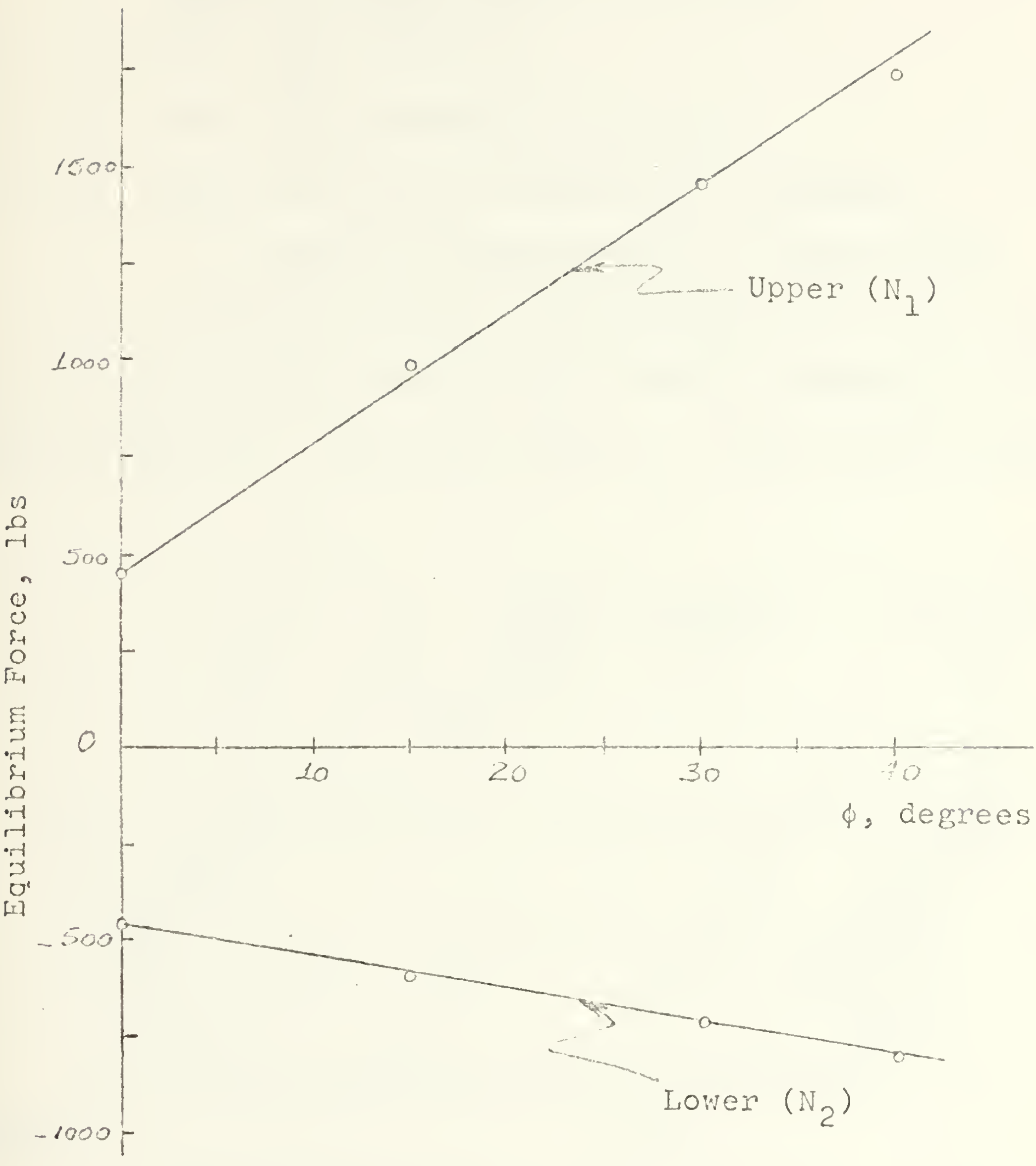


Fig. 13 Equilibrium Forces vs. Surface Inclination – Loading Condition No. 3

b. Magnetic Holding Power

Consider a free body diagram of one magnet and its supporting plate, from Fig. 43, as shown in Fig. 14(a).

The following notation was used:

R = wheel reaction

N = total structure reaction

F_H = resultant of uniformly distributed magnetic holding power, q_H, per magnet

In Fig. 14(a) $R_2 > R_1$ and $N_2 > N_1$ due to the linear variation of forces on the vertical portion of the magnetic harness. Since the distance between these forces

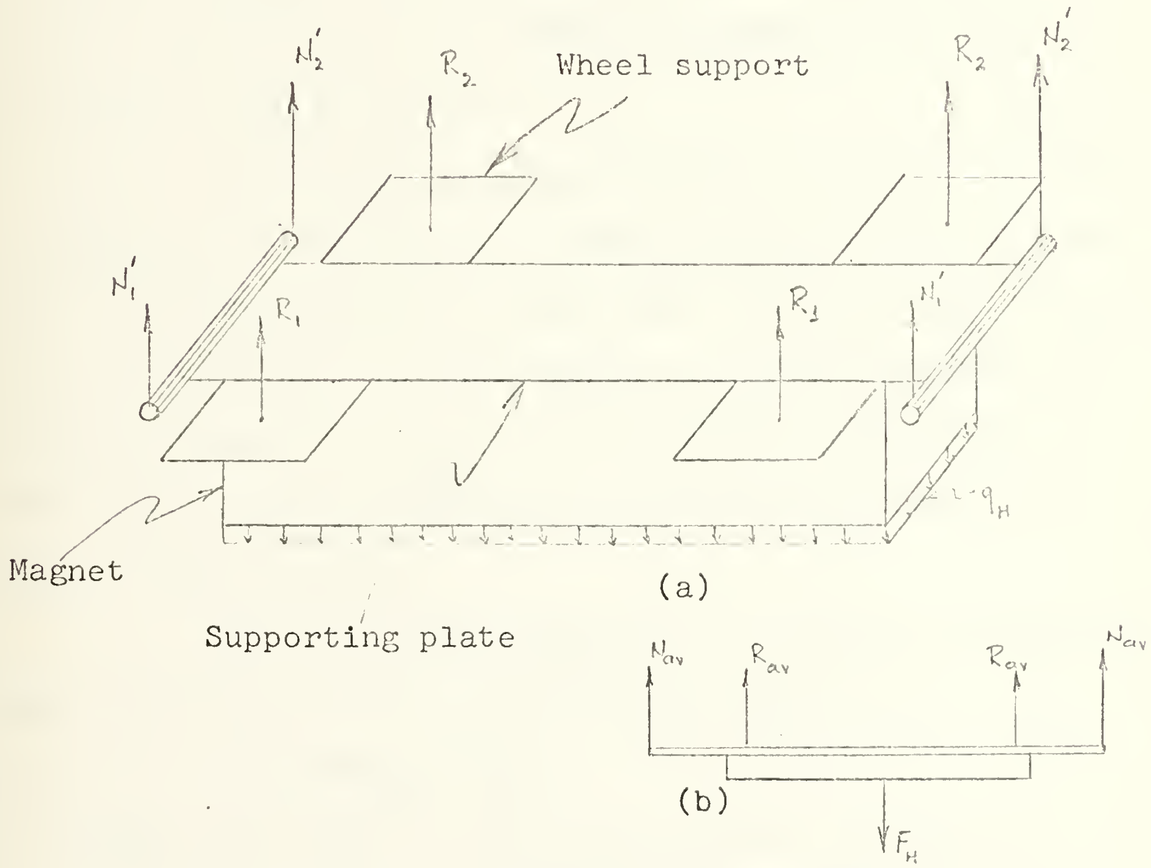


Fig. 14 Free Body Diagram of Magnet and Supporting Plate

is small in comparison with the size of the whole structure the following assumptions were made:

$$R_{av} = 2R = R_1 + R_2$$
$$N_{av} = N/4 = N_1' + N_2'$$
$$R_{min} = 25 \text{ lbs}$$

With these assumptions the analysis was based on the configuration shown in Fig. 14(b).

For the upper magnetic set, where four magnets were used (Fig. 43), two on each side, the equilibrium equation used to obtain the magnetic holding power per magnet is:

$$2F_H = 2N_{av} + 2R_{av}$$

where $N_{av} = \frac{N_1}{4}$, N_1 being obtained from Fig. 13 and also shown in Fig. 15(a)

$$\therefore \quad F_H = \frac{N_1}{4} + 2R \qquad (1)$$

The required holding power per magnet is obtained using R_{min}, on the basis of loading condition No. 3, from eq. (1) and is plotted in Fig. 15(b). For case No. 1, which is a less severe loading condition, the maximum wheel reaction is computed using eq. (1) and the F_H obtained above. This maximum wheel reaction is plotted as a function of angle in Fig. 15(c).

For the lower magnetic set, where two magnets were used (Fig. 43) the equilibrium equation is:

$$F_H = 2N_{av} + 2R_{av}$$

where $N_{av} = \frac{N_2}{2}$, N_2 being obtained from Fig. 12 and also shown in Fig. 16(a)

$$\therefore \quad F_H = \frac{N_2}{2} + 4R \qquad (2)$$

As for the upper set, the required holding power and the maximum wheel reaction were obtained and are plotted in Fig. 16(b) and Fig. 16(c) respectively.

From this analysis the required forces for one-half of the structure are:

(i) Upper set, maximum F_H = 1,200 lbs

(ii) Lower set, maximum F_H = 600 lbs

With an assumed gap of 1/8 inch, a working surface of 1/4 inch thickness, and the magnetic harness configuration shown in Fig. 43, the above forces are obtained using ERIEZ MAGNETICS NO. 4418 magnets as shown in Fig. 17 [12]. Four magnets mounted in groups of two will be needed for the upper set and two magnets will be needed for the lower set.

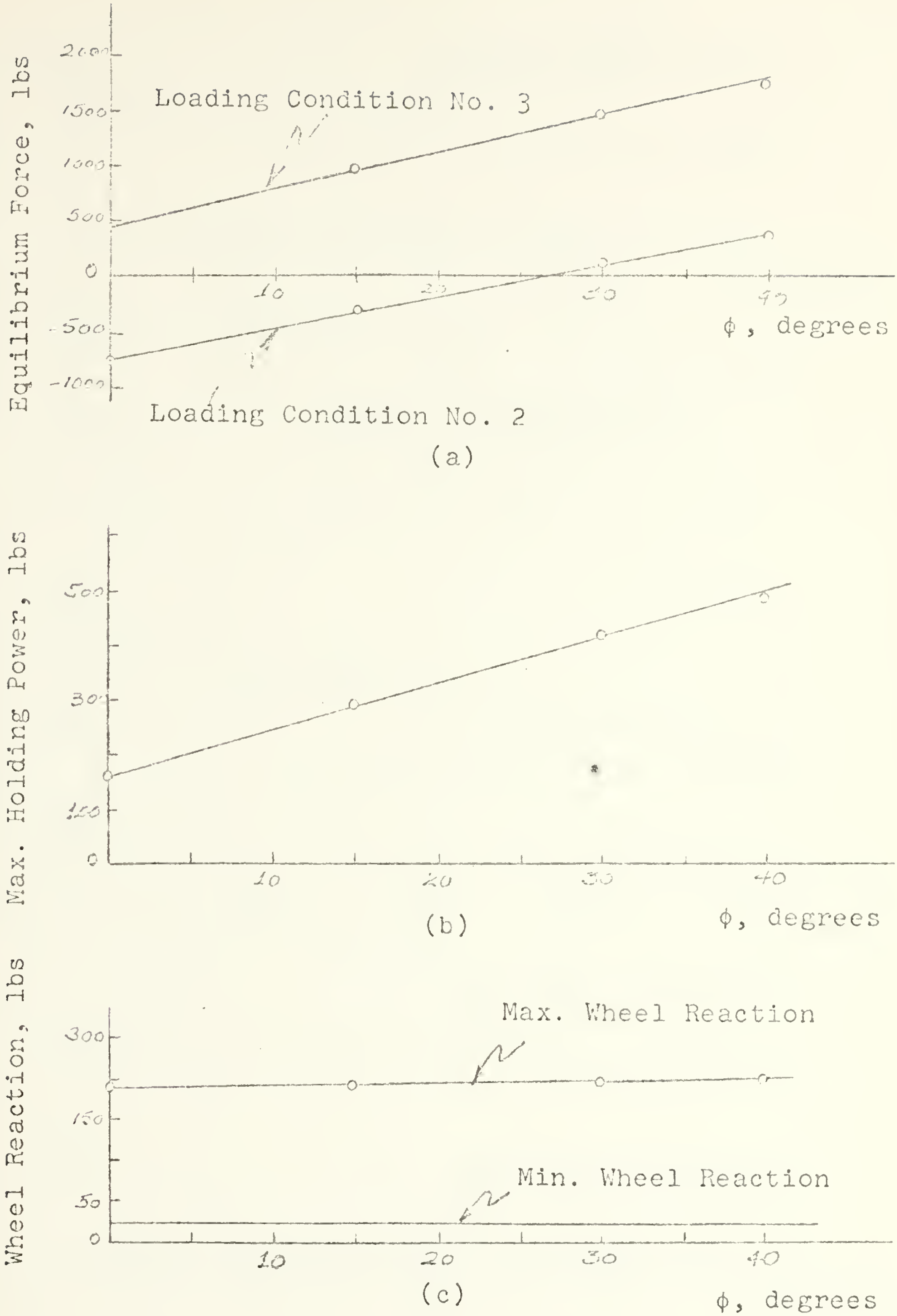


Fig. 15 Operating Conditions for Upper Magnetic Set

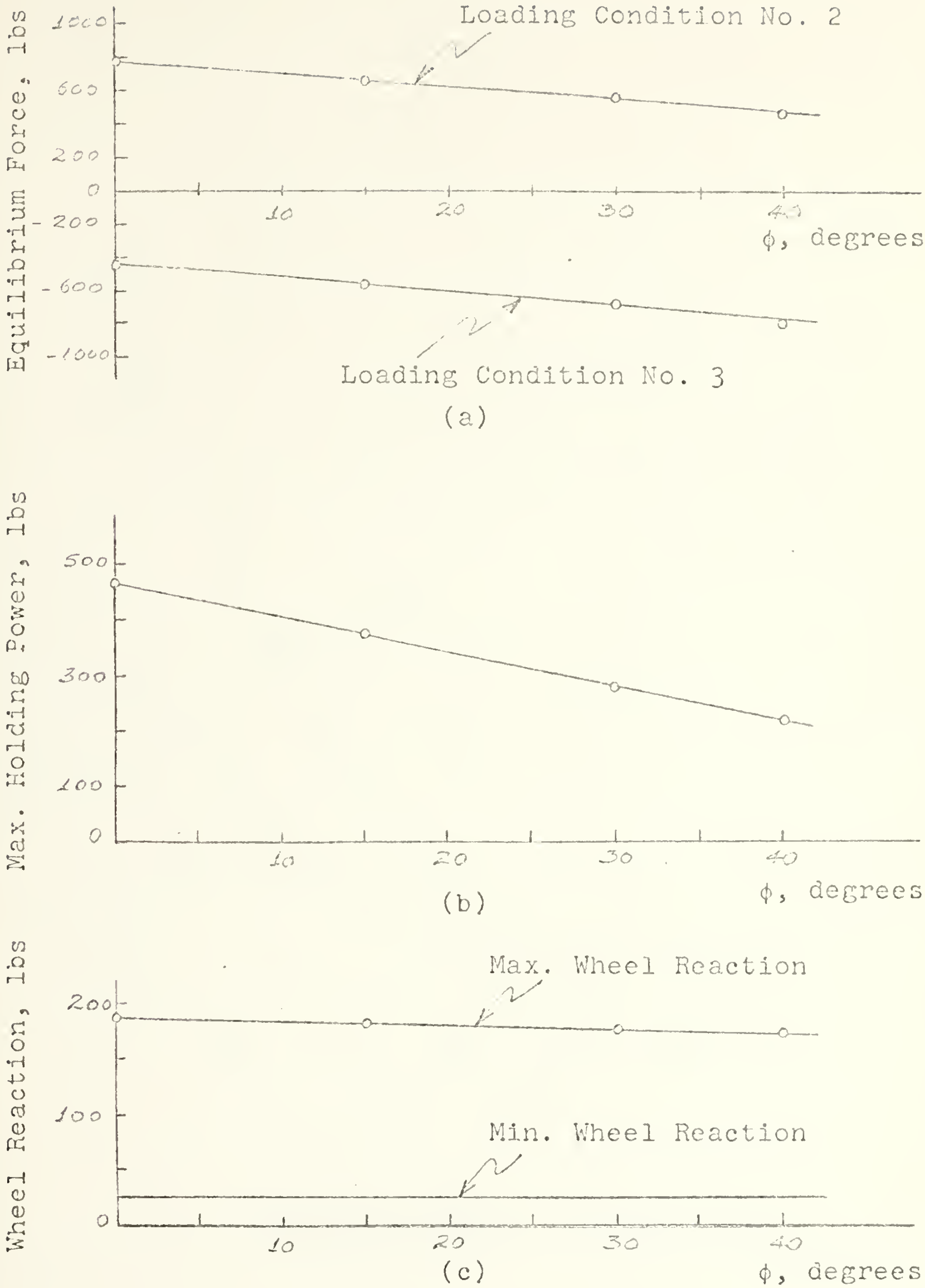


Fig. 16 Operating Conditions for Lower Magnetic Set

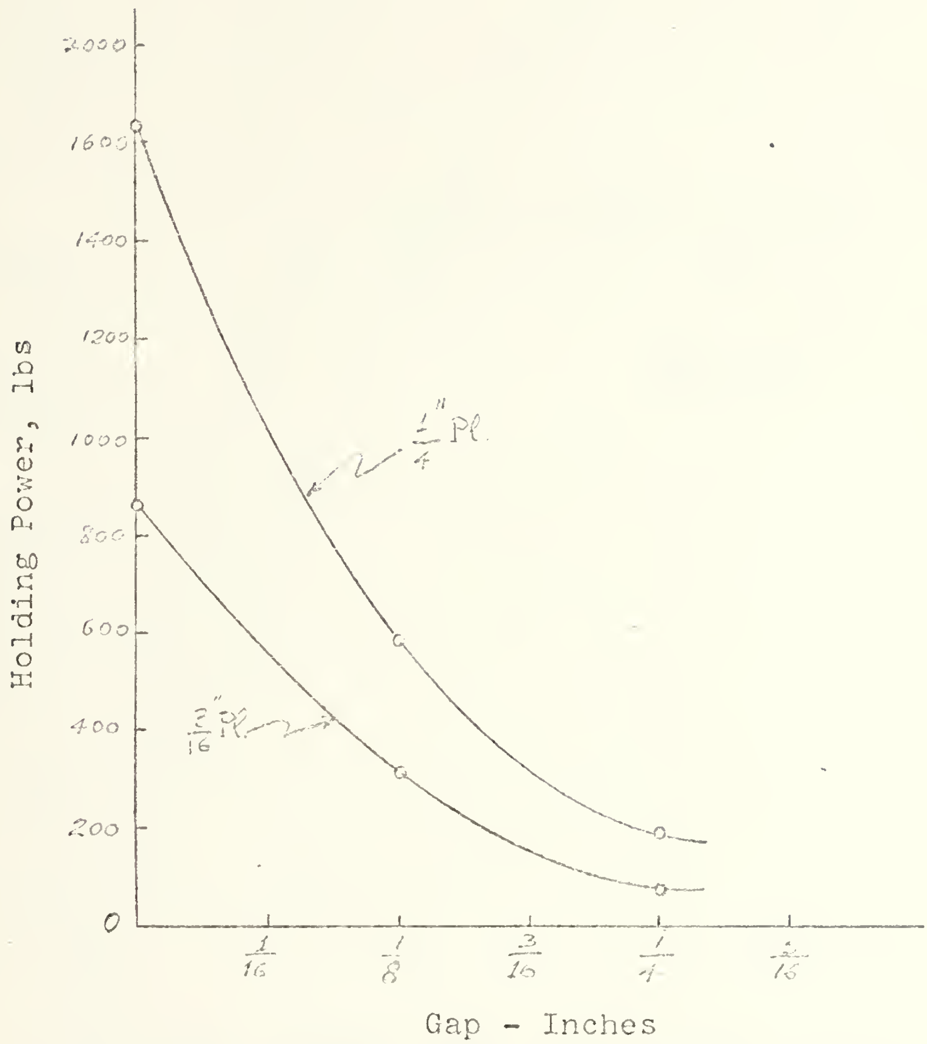


Fig. 17 Holding Power vs. Gap of ERIEZ Magnetics 4418

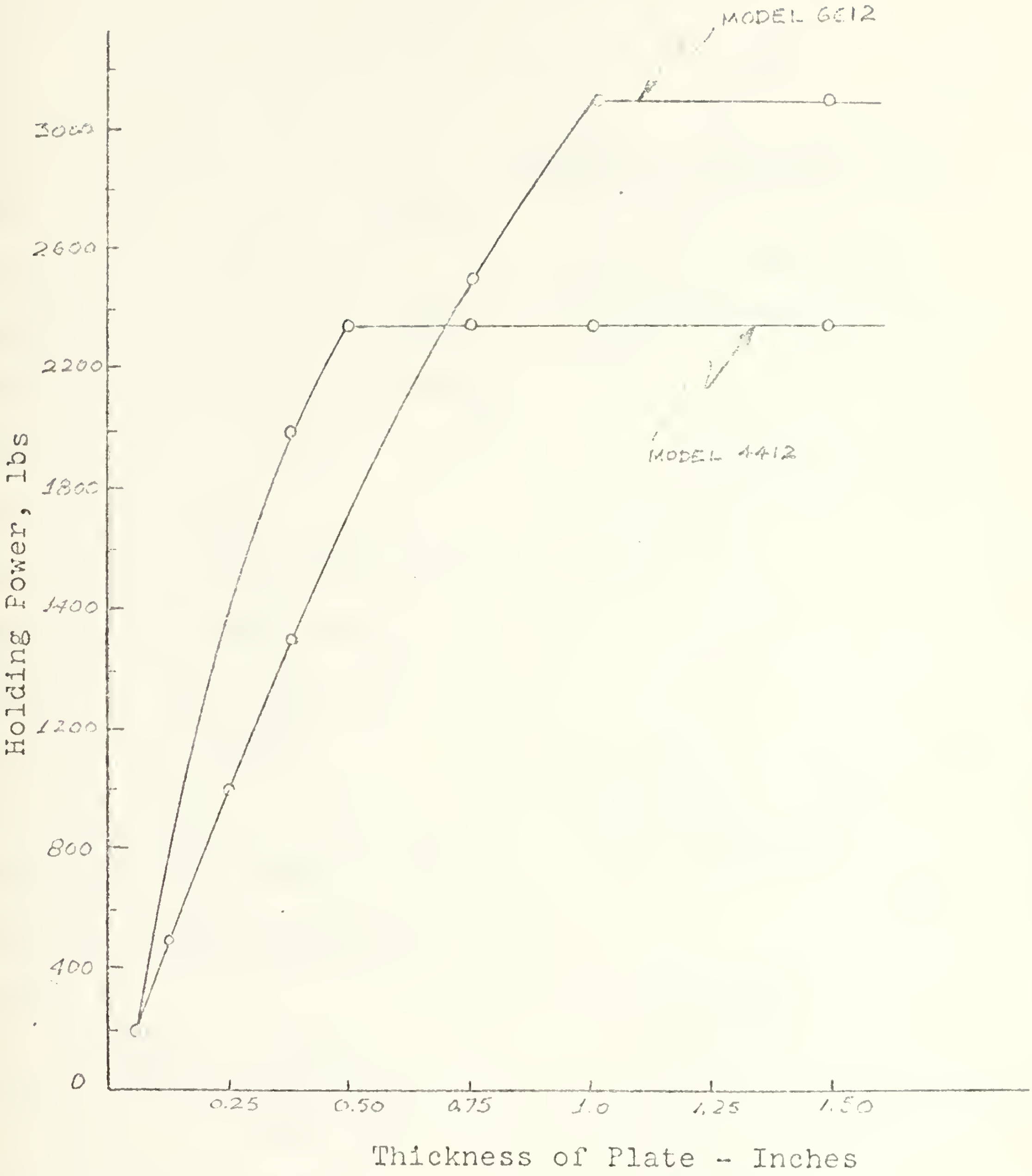


Fig. 18 Holding Power vs. Plate Thickness for Two Different Magnets

2. <u>Design of Magnetic Harness Frame</u>

Material: ASTM-A36 with mechanical properties:

σ_y = 36 ksi, E = 30 x 10^3 ksi

a. Vertical Portion

For simplicity, one of the four channels, shown in Fig. 43, was assumed to be a cantilever beam loaded as shown in Fig. 19(a). These loads occur for loading condition No. 3 at an inclination angle of 40 degrees as shown in Table No. 3. The cross-section of this beam is shown in Fig. 19(b).

I = 0.28 in^4.

Neglecting the effect of the axial force H_A the maximum stress is:

$$\sigma_{f_{max}} = \frac{M_{max}}{Z} = 14{,}500 \text{ psi at point B}$$

For allowable stress

σ_a = 36 ksi,

the factor of safety is:

$(S.F)_f$ = 2.5

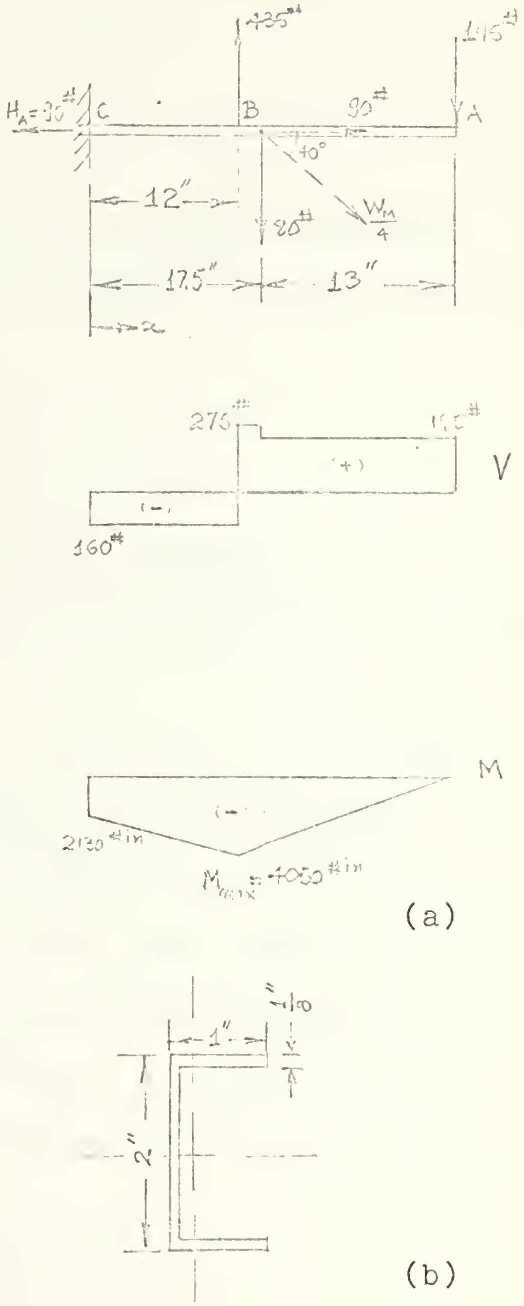


Fig. 19 Vertical Support of Magnets

The equation of the elastic curve is given as: [21].

$$EIy = -1065x^2 - 96.6x^3 + 72.5\langle x-12\rangle^3 - 13.3\langle x-17.5\rangle^3 - 32.5\langle x-30.5\rangle^3$$

at $x = 12$ in. $\quad y = 0.001$ in.

$x = 32.5$ in. $\quad y = 0.007$ in.

$\therefore$ Use 2 ⊔ 1.78

b. Horizontal Portion

A free body diagram of half the horizontal portion is shown in Fig. 20. The loads are those obtained from Fig. 19.

The following notation was used:

F_S = ball screw reaction

H_A, V_A = pivoting point reactions

From the equations of equilibrium the following were obtained:

$F_S = 330$ lbs, $H_A = 620$ lbs, $V_A = 527$ lbs

Now, consider portion AB as being fixed at one end and simply supported at the other as shown in Fig. 21(a), where the effects of the axial loads were neglected.

Bar AB is tapered and an average cross-section was assumed, as shown in Fig. 21(a). $I = 0.167$ in^4.

The maximum stress and deflection are given as:

$$\sigma_{f_{max}} = \frac{Mc}{I} = 9{,}600 \text{ psi at point B}$$

$$\delta_{max} = \frac{Pab^2}{6EI}\sqrt{\frac{a}{2\ell + a}} = 0.009 \text{ in.}$$

$$\text{at } x = \ell\sqrt{\frac{a}{2\ell + a}} = 15 \text{ in.}$$

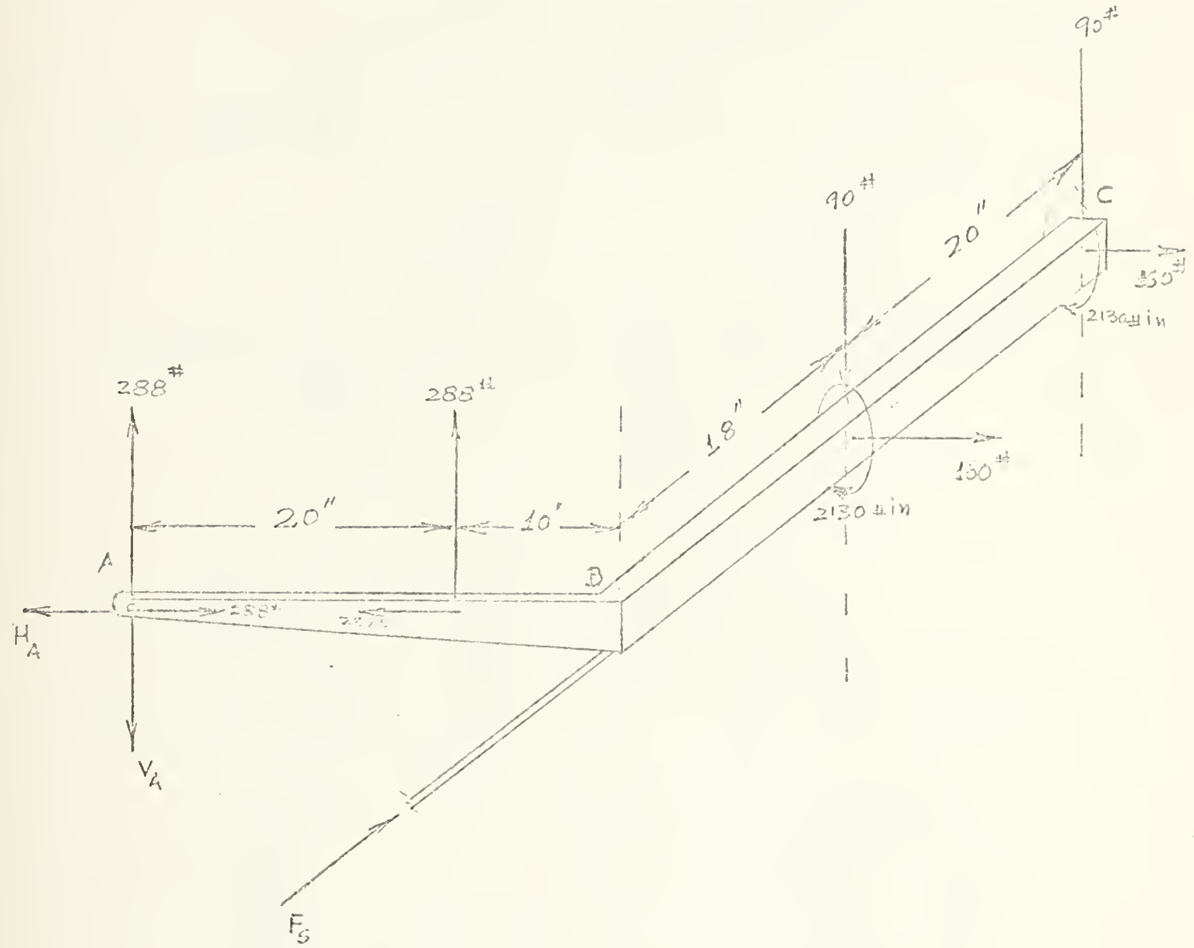


Fig. 20 Portion of Horizontal Structure of Magnetic Harness

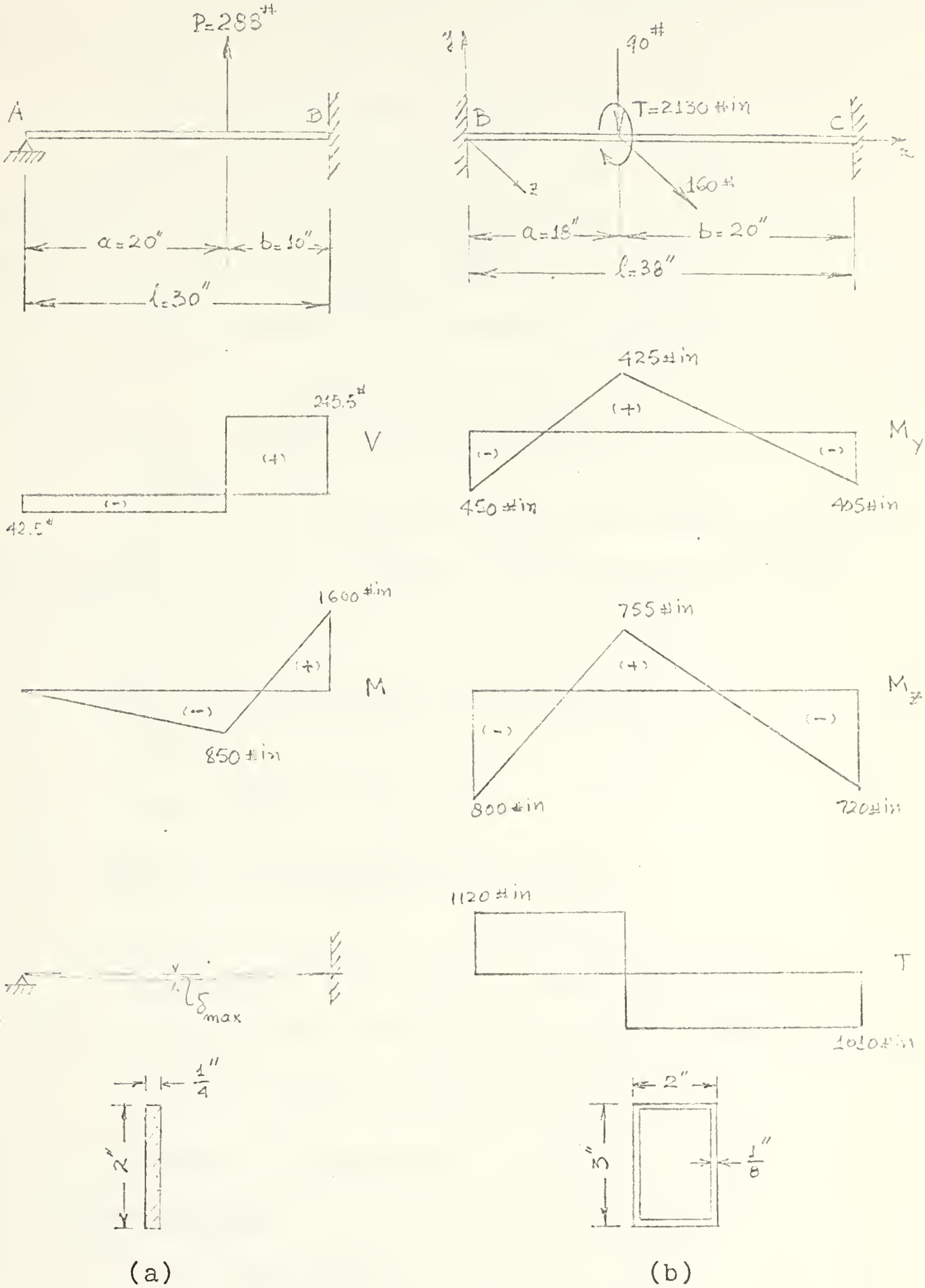


Fig. 21 Sections of Magnetic Harness Structure

For allowable stress and deflection $\sigma_a = 36$ ksi and $\delta_a = 0.03$ in. the safety factors are:

$(S.F)_f = 3.75$ and $(S.F)_\delta = 3.3$

Portion BC of Fig. 20 is represented as a built-in beam acted on by the loads and of cross-section shown in Fig. 21(b). $I_z = 1.5$ in^4, $I_y = 0.78$ in^4.

The maximum bending stress at point B is given as:

$$\sigma_{f_{max}} = \sqrt{\sigma_z^2 + \sigma_y^2} = 1{,}480 \text{ psi}$$

Neglecting the effect of direct shear, V, the maximum shearing stress at point B and the angle of twist due to torsion is given as:

$$\sigma_{s_{max}} = \frac{T}{2t(a-t)(b-t)} = 830 \text{ psi}$$

$$\theta_{max} = \frac{Ta}{KG} = 0.001 \text{ rad}$$

$$\text{where } K = \frac{2t(a-t)^2(b-t)^2}{(a+b-t)} = 1.46$$

For allowable stress and angle of twist $\sigma_a = 36$ ksi and $\theta_a = 0.003$ rad the factors of safety are:

$(S.F)_f = 20$ and $(S.F)_\theta = 3.$

C. MOVING PLATFORM CALCULATIONS

1. Force Analysis

a. Power Requirement

A free body diagram of the moving platform, shown in Fig. 47, is considered acted on by the forces shown in Fig. 22.

The following notation was used:

$F_{1,2}$ = support reactions, lbs

F_p = ship hull reaction, lbs

F_f = frictional force, lbs

T = torque on the wheel, lb-ins.

P = horse power, $\frac{T \times rpm}{63025}$

F_c = cable reaction, lbs

W = weight of structure and equipment, lbs

ϕ = angle of surface inclination, degrees

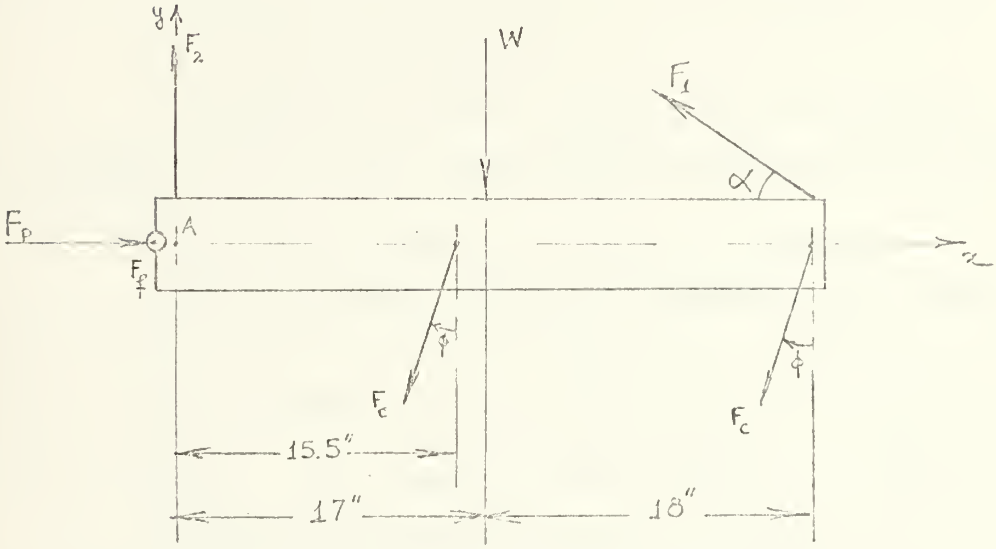


Fig. 22 Free Body Diagram of Moving Platform

The equilibrium equations are:

$$
\begin{aligned}
\Sigma F_x &: & F_p &= 2F_c \sin\phi + F_1 \cos\alpha \\
\Sigma F_y &: & F_2 + F_1 \sin\alpha &= 2F_c \cos\phi + W \qquad (1) \\
\Sigma M_A &: & 35F_1 \sin\alpha &= 17W + 50.5F_c \cos\phi
\end{aligned}
$$

For $W = 500$ lbs and $\phi = 30°$ equations (1) give:

$$F_1 = 486 + 2.86\ F_c\cos\phi$$

$$F_2 = 0.57\ F_c\cos\phi + 257 \qquad (2)$$

$$F_p = (2.48\ \cos\phi + 2\ \sin\phi)F_c + 422$$

For a rolling coefficient of friction $\mu = 0.05$ the maximum frictional force is:

$$F_f = \mu F_p \qquad (3)$$

Among the loading and working conditions shown in Tables 1, 2, 3, the following condition was considered as the worst:

$$\phi = 0, \quad F_c = 750 \text{ lbs}$$

For maximum linear velocity v = 30 ft/min. and wheel diameter D_w = 9 in. the maximum revolution per minute is: rpm = 13.

For the above condition equations (2) and (3) give:

F_1 = 2,626 lbs, F_2 = 686 lbs, F_p = 2,282 lbs

F_f = 114 lbs, T = 513 lb-in. P = 0.105 hp

For intermittent operation and moderate shock a service factor of 2 was used. Therefore the maximum horsepower and torque required is:

P_{max} = 0.21 hp T_{max} = 1,026 lb-ins.

Since two speed reducers are used, one at each wheel, the following were selected:

Morse Speed Reducer 18GCDV (2 REQ'D)

Output: max. rpm 14.0 Input: rpm 1750
max. torque 554 lb-ins.

Morse Electric Motor, 1/4 hp., 25TE115 (2 REQ'D)

Morse Remote Control MA-25

b. Support Design

Four supporting bars are used for the suspension of the moving platform. For each inclined bar $F_1 = 1,313$ lbs and for each vertical bar $F_2 = 342$ lbs. These supports are shown in Fig. 23(a).

Assuming the cross section of each bar as shown in Fig. 23(b), the max. stress is:

$$\sigma_{t_{max}} = \frac{F}{A} = 11,920 \text{ psi}$$

For an allowable strength $\sigma_a = 40$ ksi the safety factor is:

$$(S.F)_f = 3.35$$

∴ Use 3/8 in. hot rolled rounds (AISI 1018)

c. Wheel Arrangement

The detailed wheel arrangement is shown in Fig. 50. The lateral load is 1,141 lbs for each of the wheels as shown in Fig. 25. Each wheel is supported by two roller bearings. The following were selected: BASSICK wheel No. WR-1248 molded with a polyurethane tread or a similar material of thickness 1/2 in. (2 REQ'D). SKF Roller bearing F-47, No. 478204-012 (4 REQ'D).

2. Design of Base

Material: ASTM-A36 with mechanical properties:

$\sigma_y = 36$ ksi, $E = 30 \times 10^3$ ksi

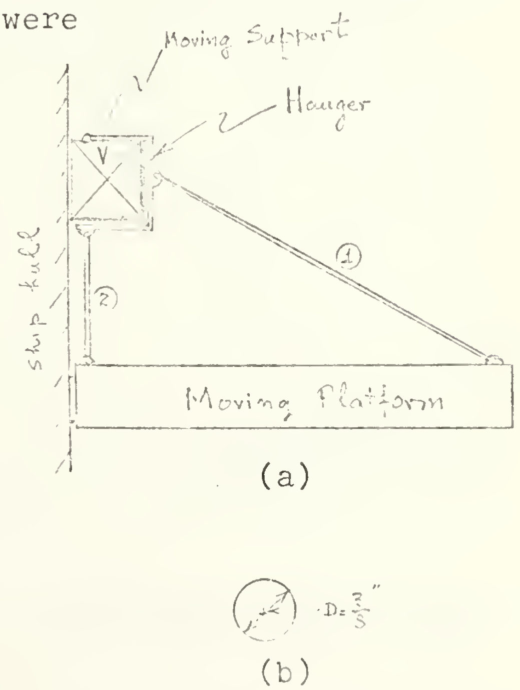


Fig. 23 Supports of Moving Platform

a. Design of Base to Resist Bending

Consider the base as a simply supported beam as shown in Fig. 24(a) where the weight of the base was neglected.

For a stiff platform base, a maximum unit deflection of 0.0001 inch per inch was chosen as a design criterion.

The following notation was used:

W_2 = 50 lbs (speed reducer)
W_1 = 200 lbs (winch)
F_c = 750 lbs (cable tension)
δ_a = 0.0001 in/in.

The equation of elastic curve is given by: [21]

$$EIy = 25x^3 - 8.4\langle x-5\rangle^3 - 33.3\langle x-37\rangle^3 - 8.4\langle x-64\rangle^3 - 8.22\cdot 10^4 x$$

at x = 37 in. $y = \dfrac{-20\cdot 10^5}{EI}$

for $y = \delta_a$ the minimum required moment of inertia

$$I_{min} = 9 \text{ in}^4$$

The cross-section shown in Fig. 24(b) is adequate, giving $I = 11 \text{ in}^4$.

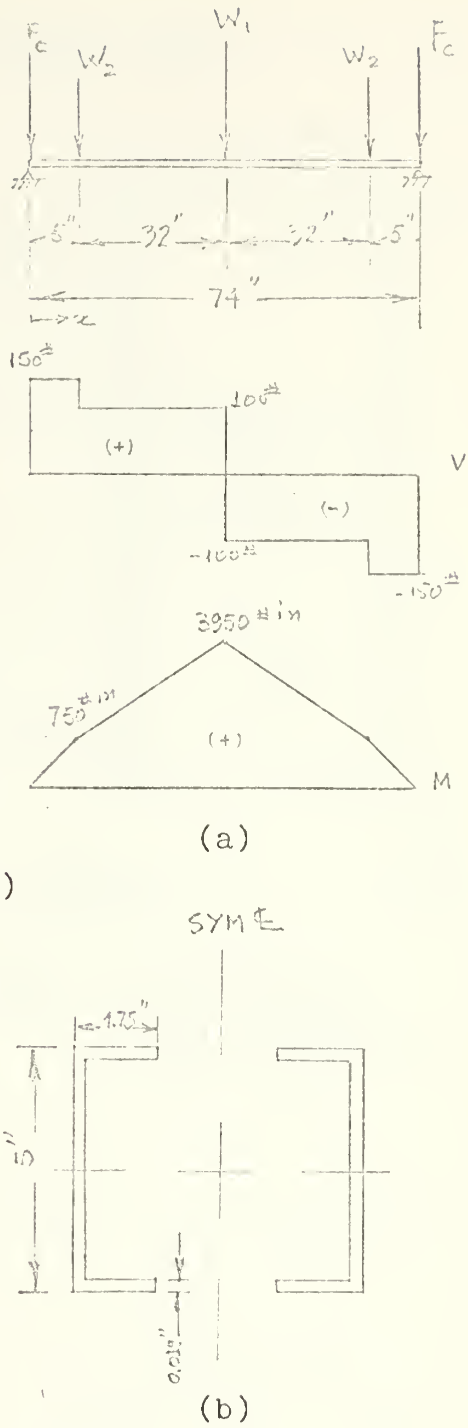


Fig. 24 Distribution of Loads on Moving Platform

The maximum stress at the middle of the beam:

$$\sigma_{f_{max}} = \frac{Mc}{I} = 900 \text{ psi}$$

b. Design Base to Resist Lateral Load

A model of the base is shown in Fig. 25, acted on by a load of 1,141 lbs on each wheel as indicated by the force analysis of Fig. 22. The base consists of:

Two channels for the support of the speed reducers and the suspension of the wheel "boxes."

Three channels completing the outside frame.

Two transverse angles supporting the winch.

The strength analysis due to these lateral loads revealed a stiff structure.

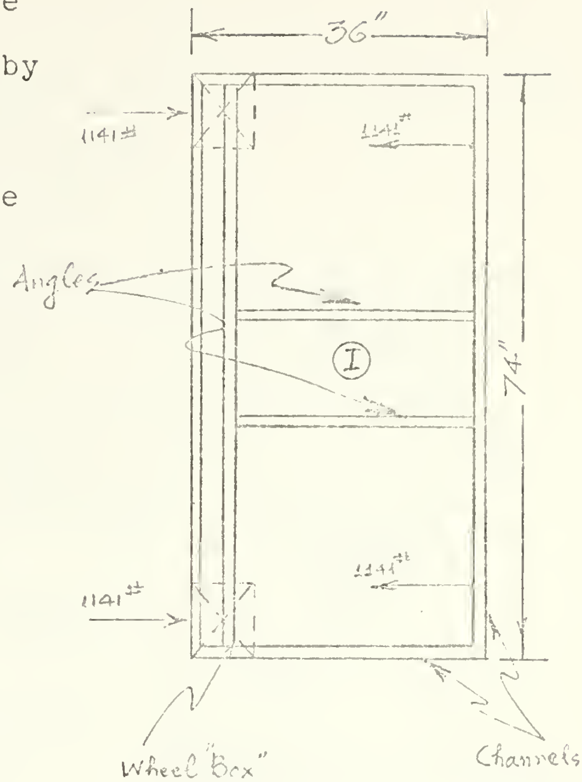
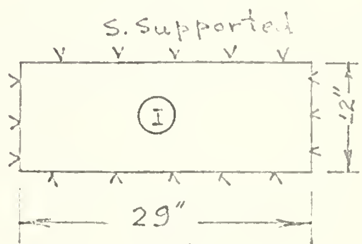


Fig. 25 Transverse Stiffeners of Base of Moving Platform

c. Design of Top Panel to Resist Localized Bending

Consider portion I of the top plate shown in Fig. 25 as 1/8 in. plate simply supported at all edges with a uniformly distributed load of $w = 0.57$ lbs/in^2 as shown in Fig. 26.

The maximum stress and deflection are given as:



(b)

Fig. 26 Portion of Top Panel of Base

$\sigma_{f_{max}} = \beta \frac{wb^2}{t^2} = 3,460$ psi, $\delta_{max} = \alpha \frac{wb^4}{Et^3} = 0.24$ in. at center of plate.

For allowable stress σ_a = 36 ksi and deflection δ_a = 0.030 in. the safety factors are:

$(S.F)_f = 10.4$ and $(S.F)_\delta = 1.4$

∴ Use: 5⊔6.7, ∟3 x 3 x 1/4, 1/8 in. diamond plate

D. MOVING SUPPORT CALCULATIONS

The moving support consists of the structure and the hanger as shown in Figs. 51, 55 respectively.

Material for structure: Steel AISI M-1020 with mechanical properties;

σ_y = 35 ksi, E = 30 x 10^3 ksi

1. Design of Hanger

a. Connecting Axle

Consider a free body diagram of the hanger as shown in Fig. 27(a). The axle through point A is shown in Fig. 27(b), acted on by the loads due to suspension of the moving platform. The cross-section is also shown in Fig. 27(c). I = 0.00306 in^4.

The combined maximum bending moment and stress are given as:

$M_{max} = \sqrt{M_x^2 + M_z^2}$ = 570 lb-ins

$\sigma_{f_{max}} = \frac{Mc}{I}$ = 46,490 psi at an

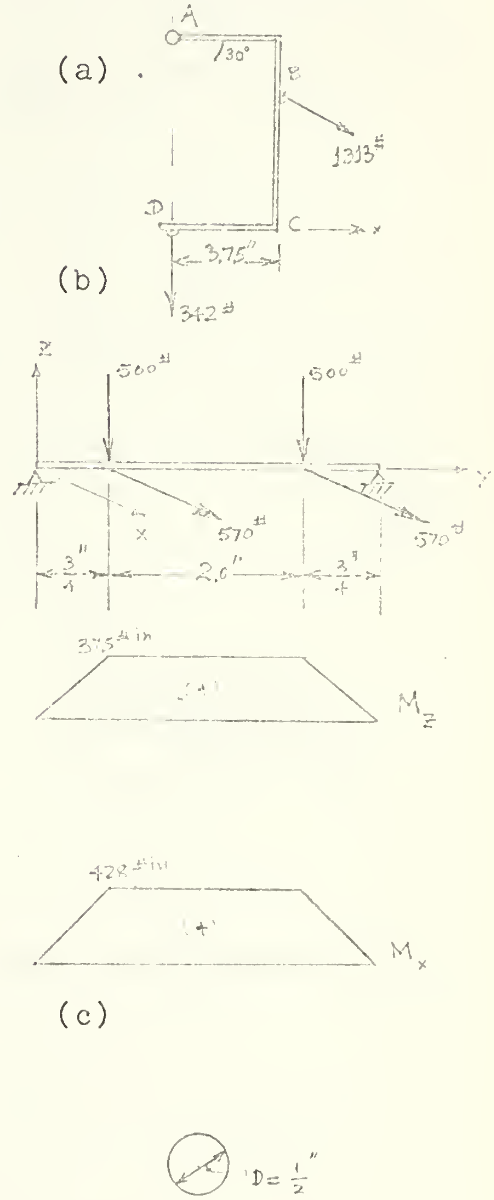


Fig. 27 Platform Hanger Support

angle of 49 degrees with respect to the horizontal plane.

The maximum combined deflection is:

δ_{max} = 0.003 in.

For allowable stress σ_a = 97 ksi and deflection δ_a = 0.03 in. the factor of safety:

$(S.F)_f$ = 2.1 and $(S.F)_\delta$ = 10

∴ Use 1/2" hot rolled carbon steel bar (AISI 1144).

b. Hanger Structure

Consider portion CD of Fig. 27(a) as being a cantilever beam of cross-section (I = 0.17 in^4) and loading as shown in Fig. 28.

The maximum stress and deflection are:

$\sigma_{f_{max}}$ = 7,000 psi

δ_{max} = 0.002 in.

For allowable stress and deflection σ_a = 35 ksi and δ_a = 0.03 in. the safety factors are:

$(S.F)_f$ = 5 and $(S.F)_\delta$ = 15

∴ Use 1/4 in. hot rolled flat (AISI M-1020)

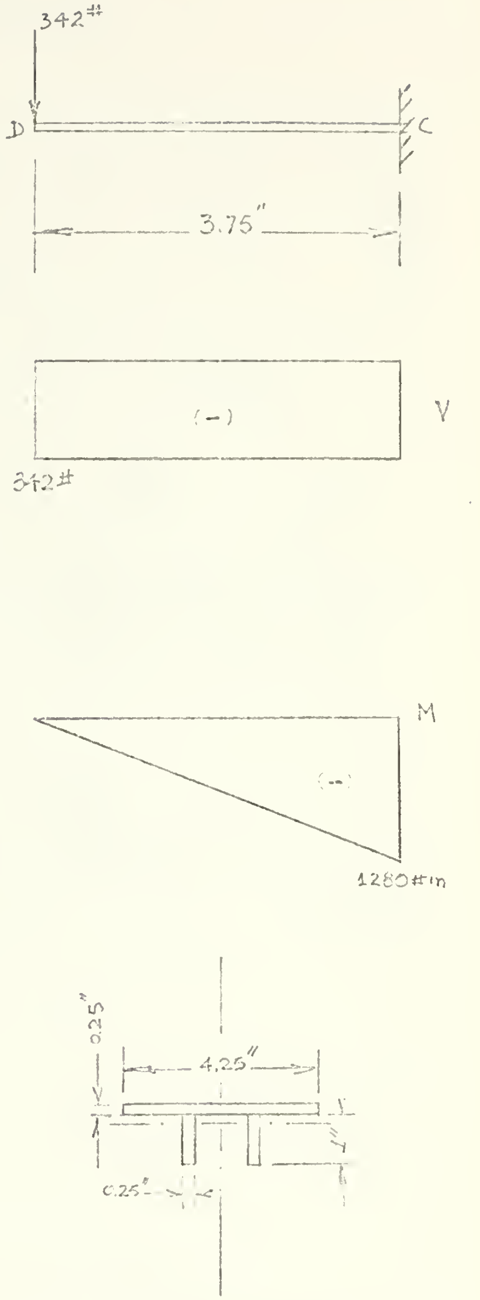


Fig. 28 Hanger Structure

2. Design of Moving Support Structure

A free body diagram of the middle horizontal plate of the moving support structure shown in Fig. 52 is considered as 1/8 in. plate simply supported at two edges and free at the others as shown in Fig. 29(a). The acting loads are those shown in Fig. 27(a). For simplicity assume the shaded portion ABCD as a simply supported beam acted on by concentrated loads shown in Fig. 29(b) and having a cross-section shown in Fig. 29(c).

$$I = 0.0026 \text{ in}^4.$$

The maximum stress and deflection are:

$\sigma_{f_{max}} = 18{,}000$ psi

$\delta_{max} = 0.005$ in..

For allowable stress and deflection $\sigma_a = 35$ ksi and $\delta_a = 0.03$ in. the factors of safety are:

$(S.F)_f = 1.95$ and $(S.F)_\delta = 6$

$\therefore$ Use 1/8 in. hot rolled sheet (M-1020).

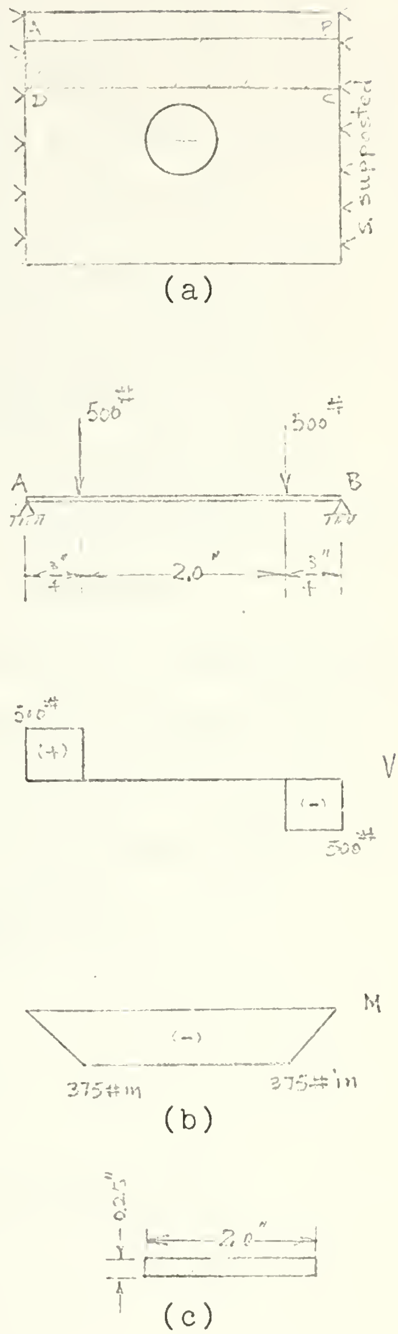


Fig. 29 Upper Plate of Moving Support Structure

E. EXTERNAL SUPPORT CALCULATIONS

Material: Steel C-1045 with mechanical properties:

$$\sigma_y = 57 \text{ ksi}, \quad E = 30 \times 10^3 \text{ ksi}$$

A free body diagram of portion of the external support shown in Fig. 32 is considered in Fig. 30. The loads acted on by the wheels of one of the moving supports are uniformly distributed and the constraints were assumed as, edge AF fixed, the others free. The evaluation of the deflection and stress distribution over the entire plate of this configuration requires numerical techniques or solution of a fourth order differential equation.

For simplicity it is assumed all loads are concentrated and acting simultaneously on the beam shown in Fig. 31(a). The cross-section of the beam and force analysis is shown in Fig. 31(b), (c), (d) respectively. ($I = 0.104 \text{ in}^4$)

The stress distribution and deflection are:

σ_f = 16,200 psi at point B

$\sigma_{f_{max}}$ = 21,900 psi at fixed point A

δ_{max} = 0.0717 in. at point B

For allowable stress σ_a = 57 ksi and deflection δ_a = 0.1 the factors of safety are:

$(S.F)_f = 2.6$ and $(S.F)_\delta = 1.4$

$\therefore$ Use 1/2 in. plate (C-1045)

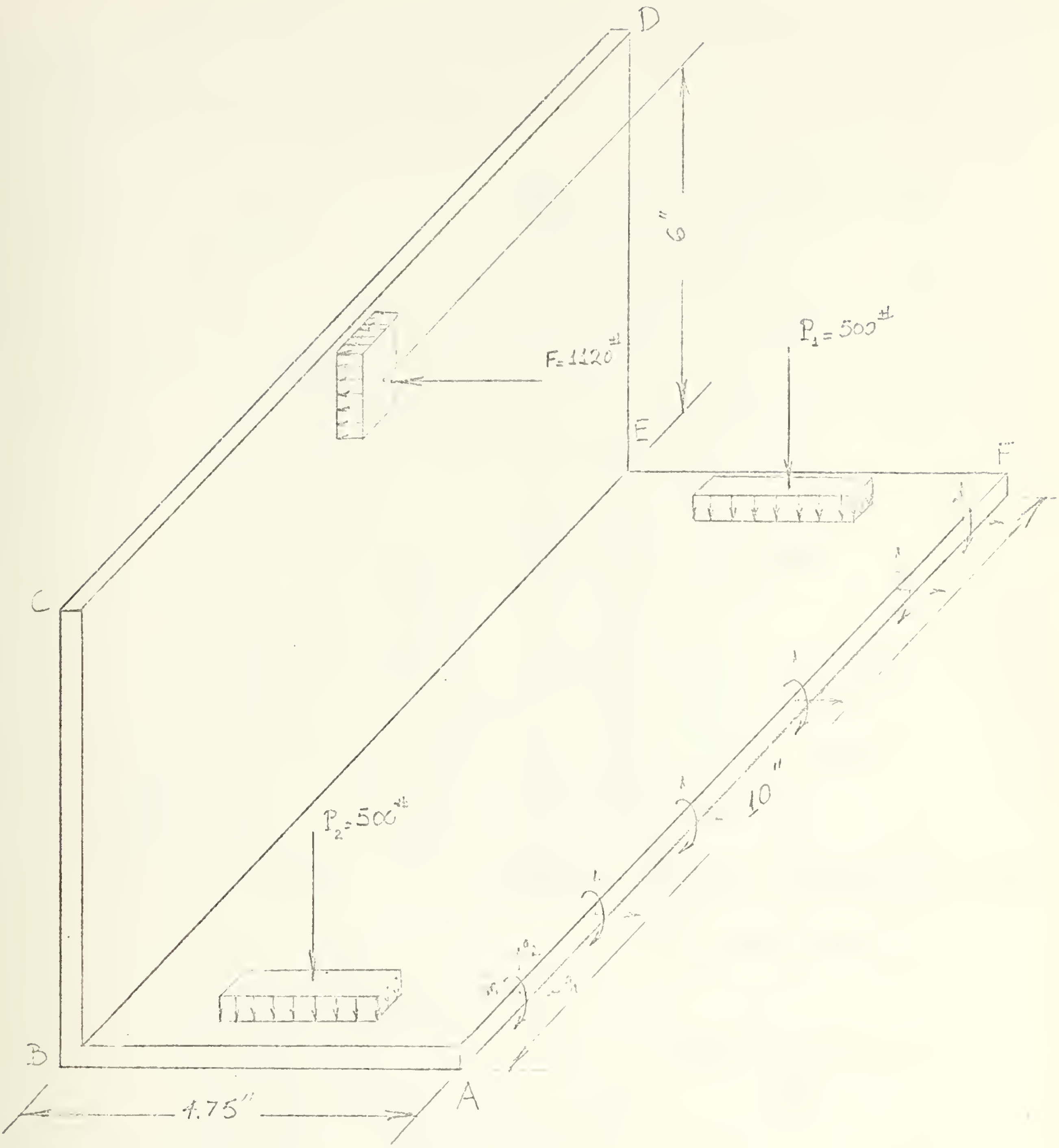


Fig. 30 Free Body Diagram of External Support

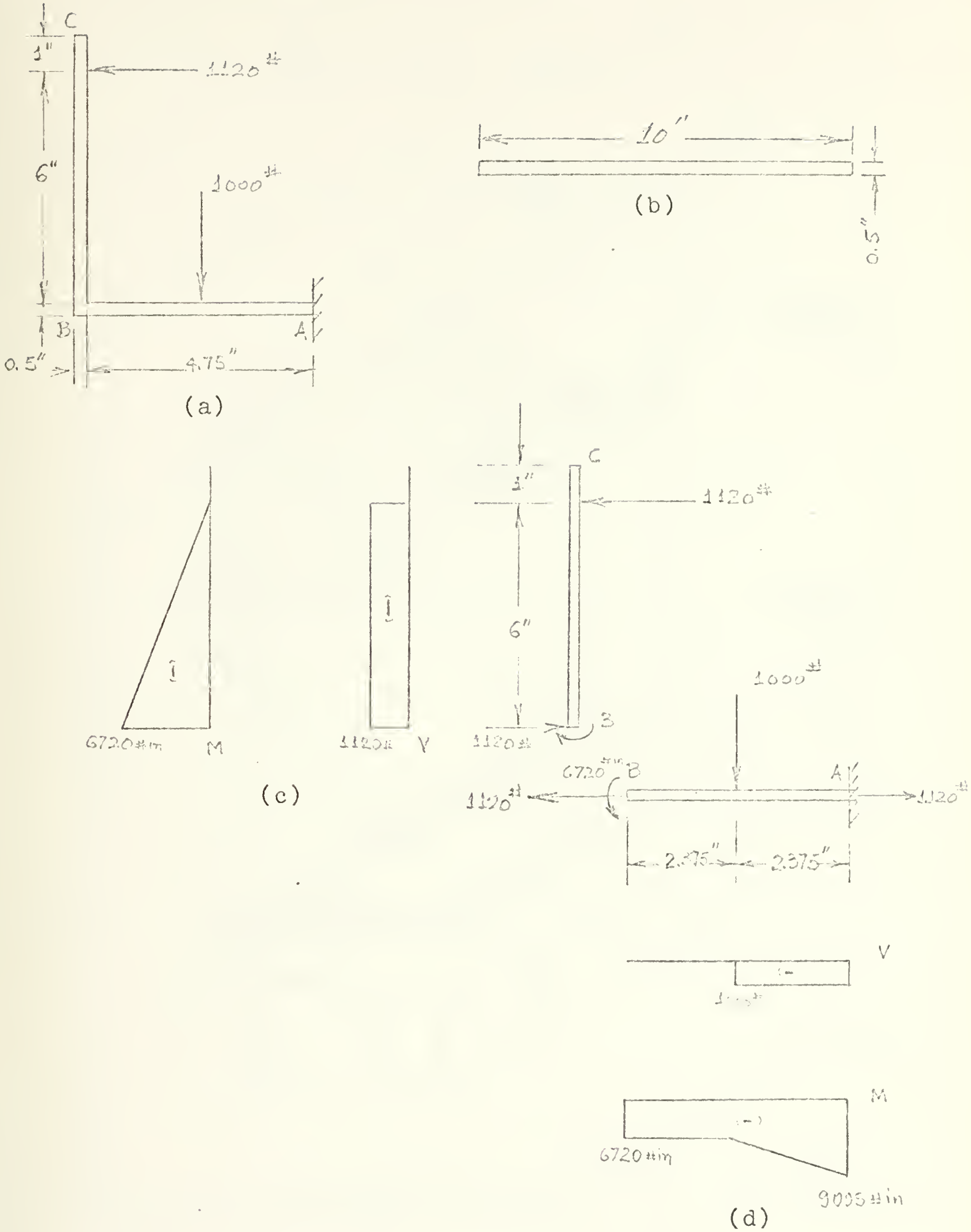


Fig. 31 Force Analysis of External Support

APPENDIX B - DETAIL DRAWINGS

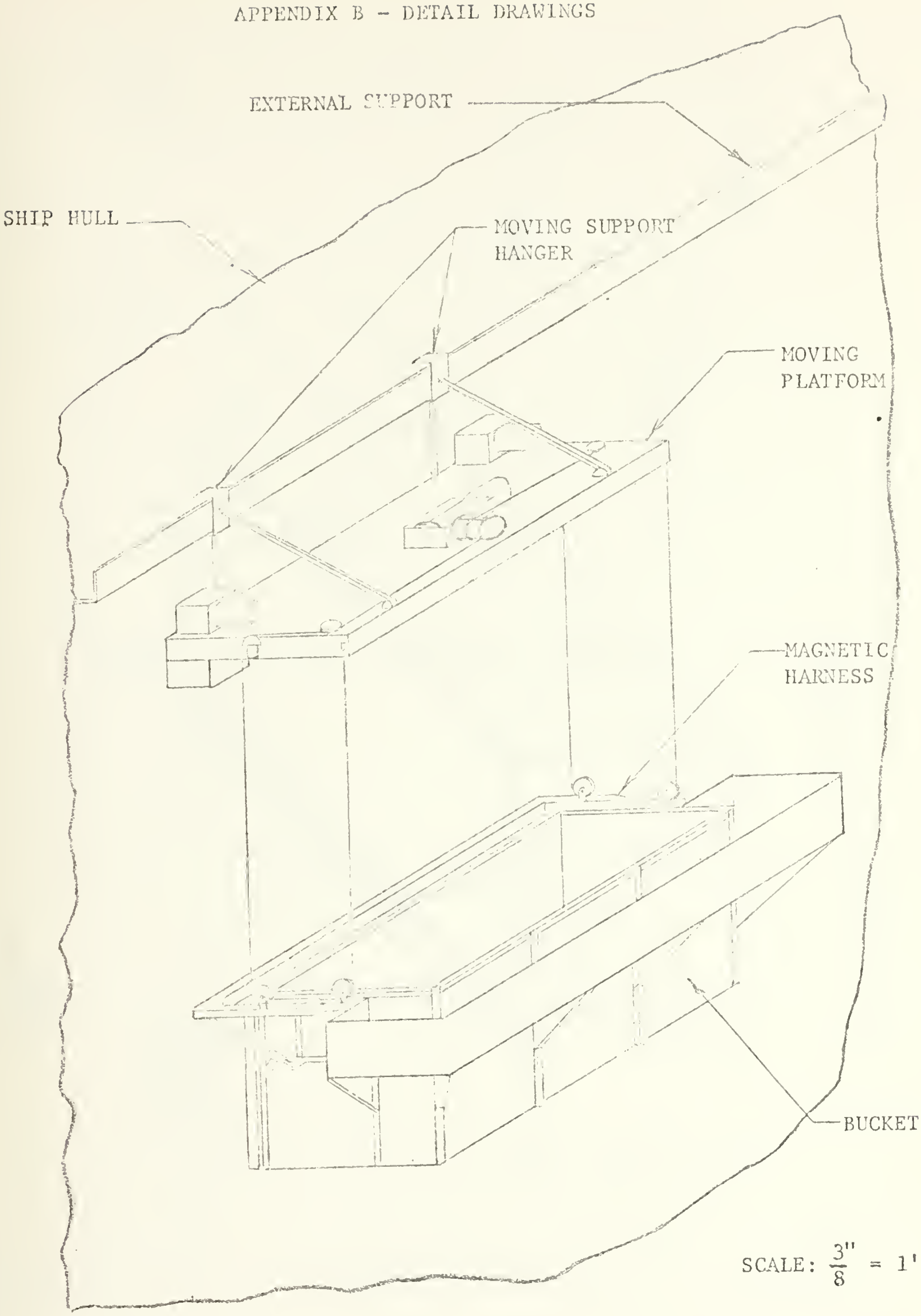


Fig. 32 General View of Self-Propelled Work Platform

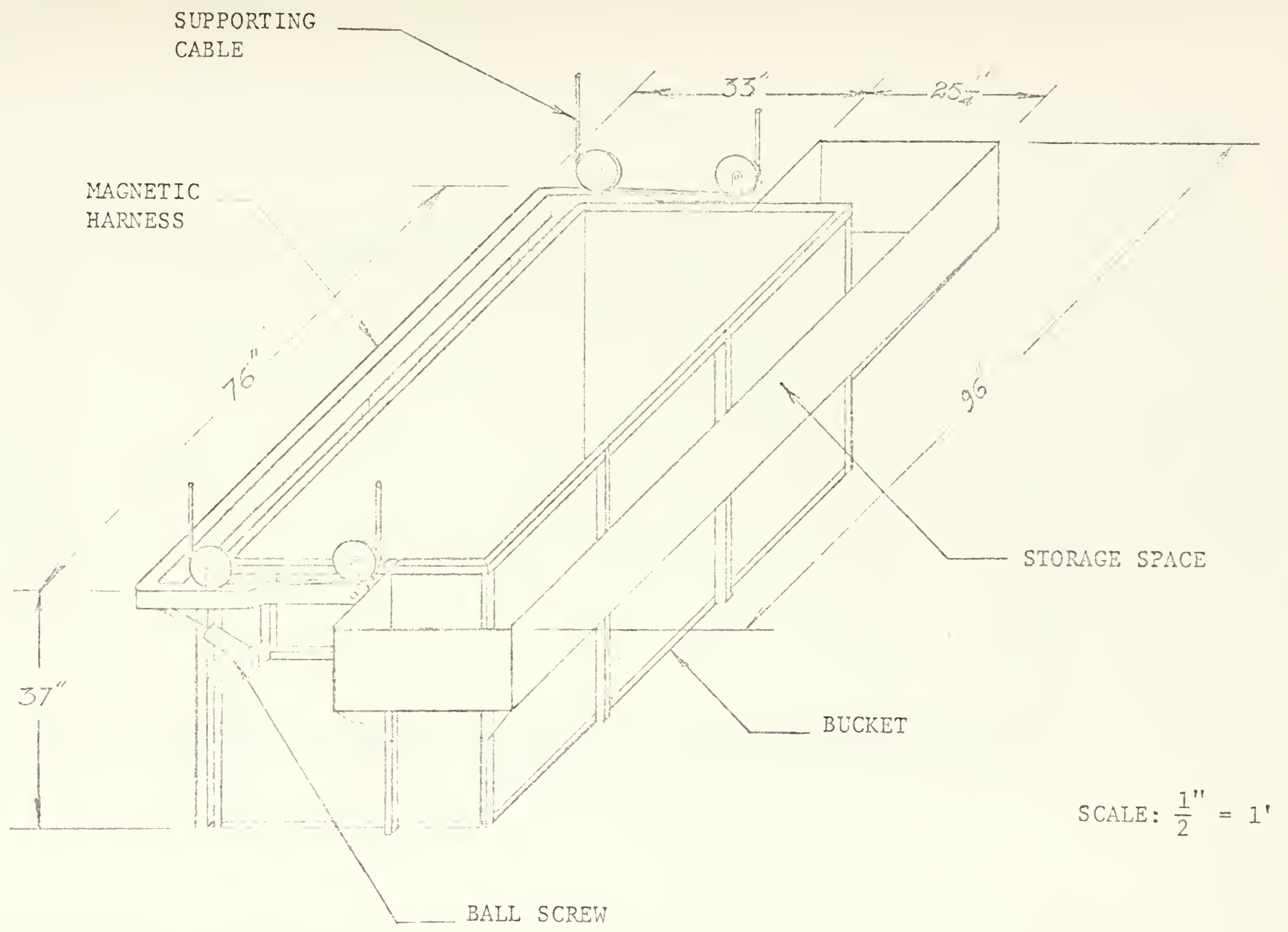
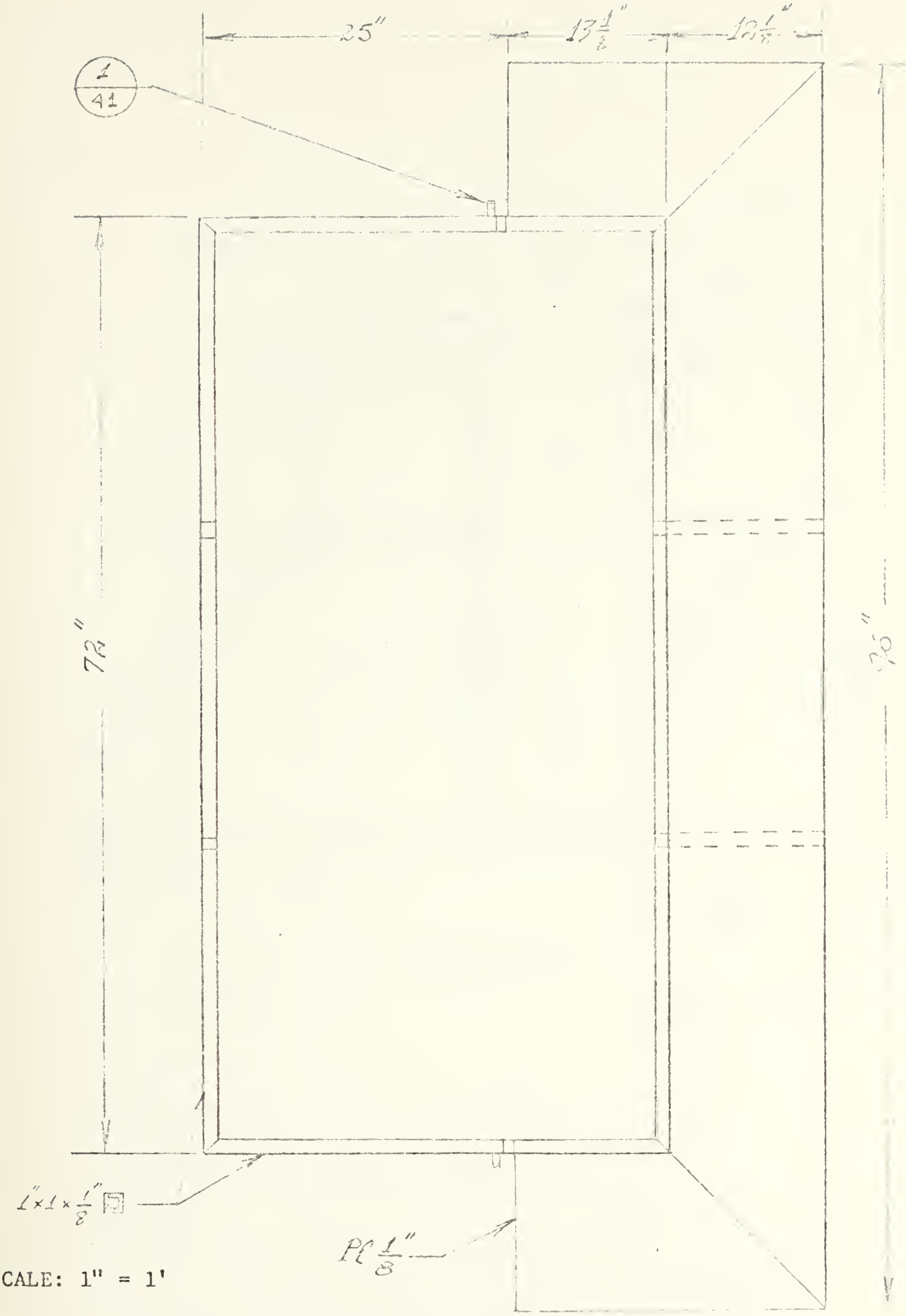
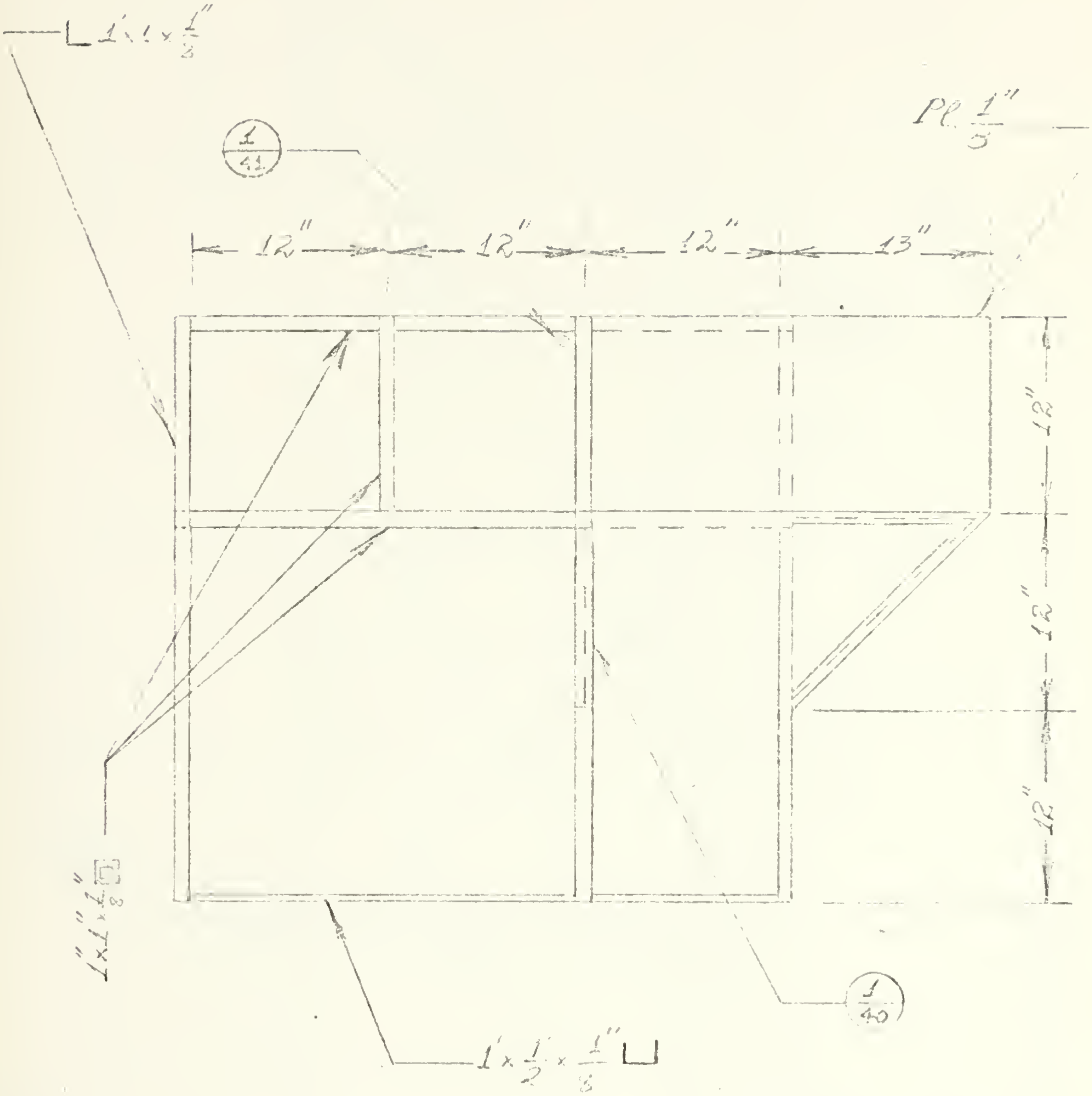


Fig. 33 General View of Bucket and Magnetic Harness



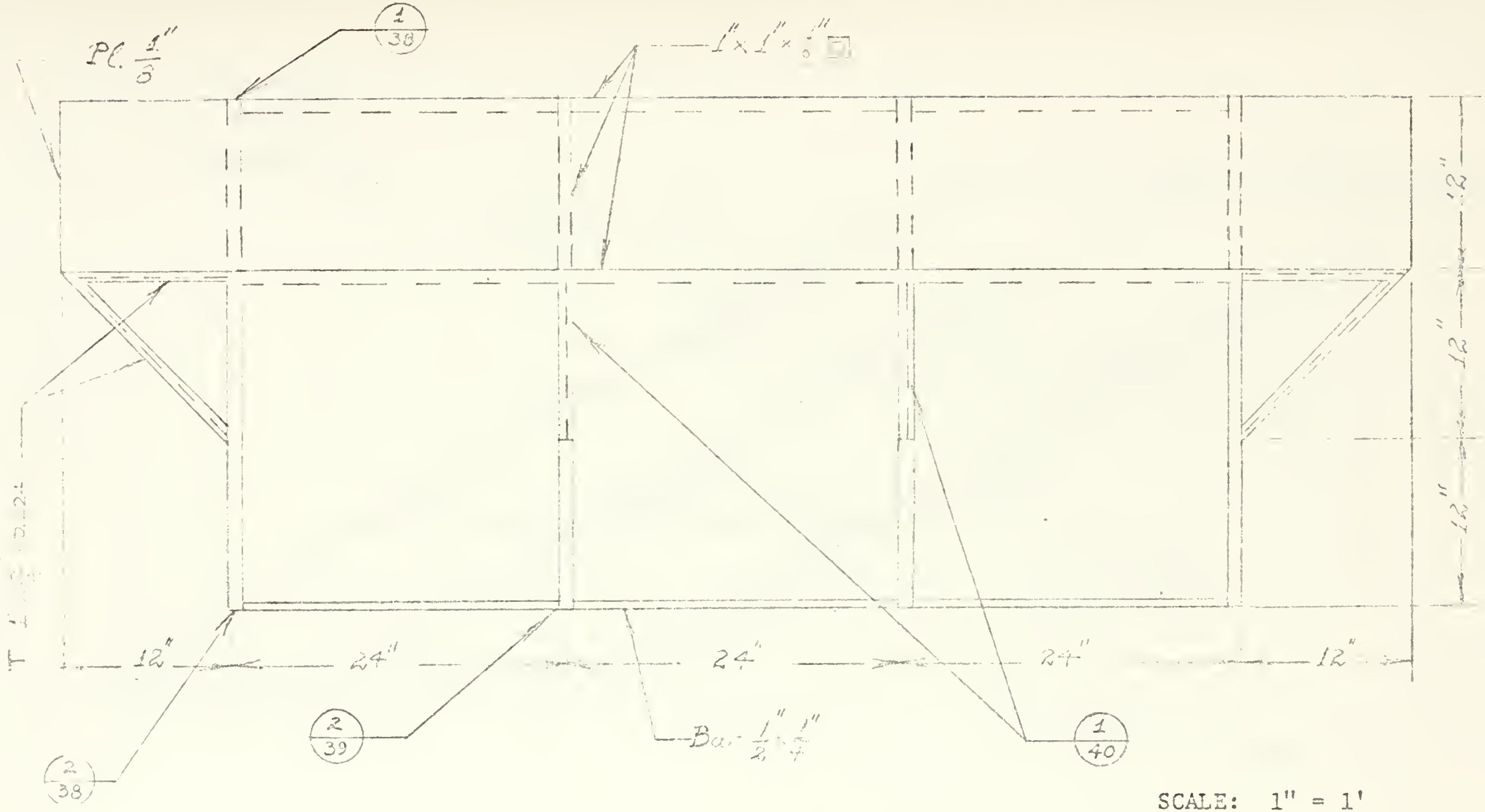
SCALE: 1" = 1'

Fig. 34 Top View of Bucket



SCALE: 1" = 1'

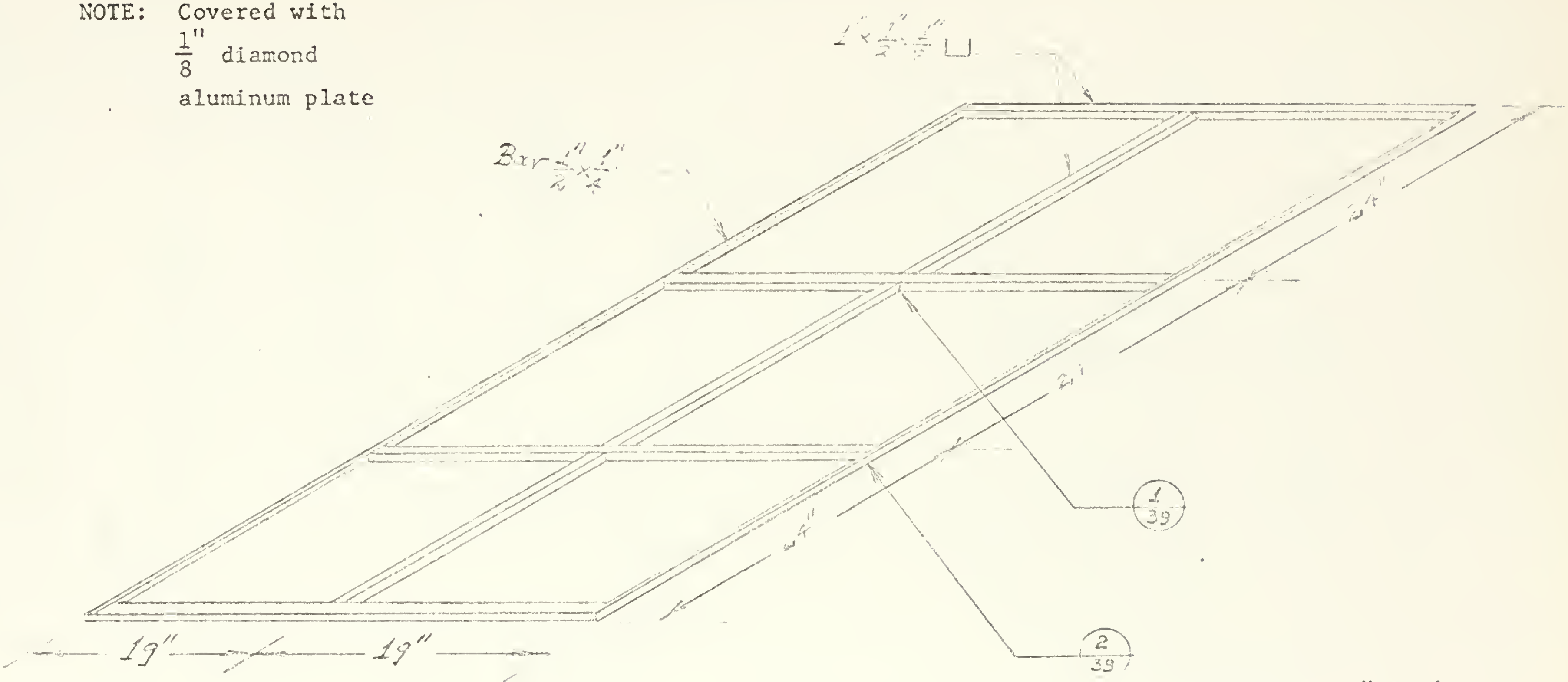
Fig. 35 Side View of Bucket



SCALE: 1" = 1'

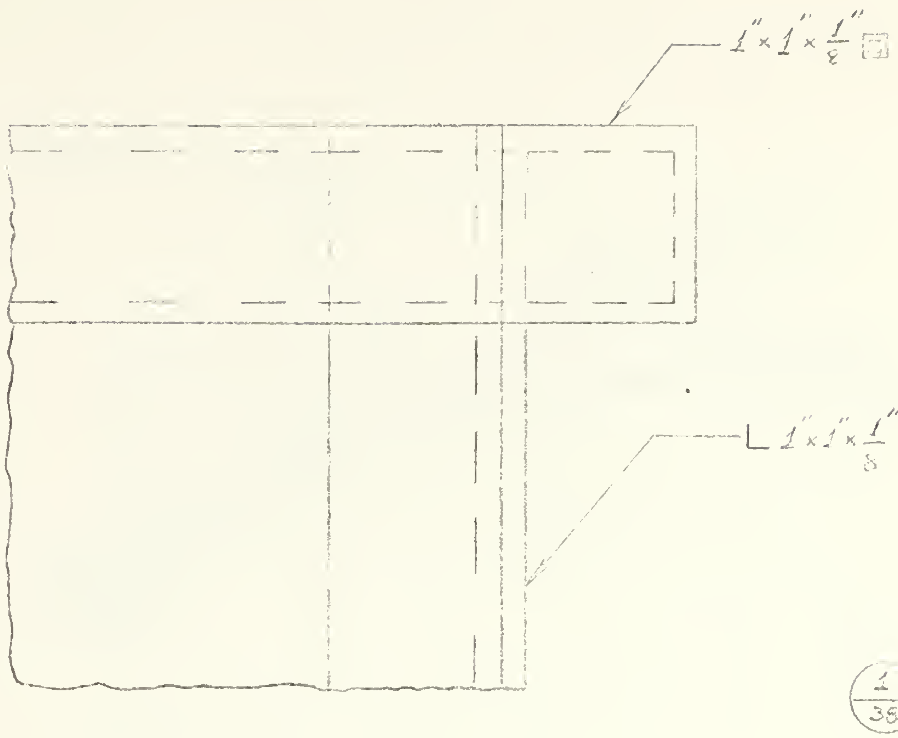
Fig. 36 Back View of Bucket

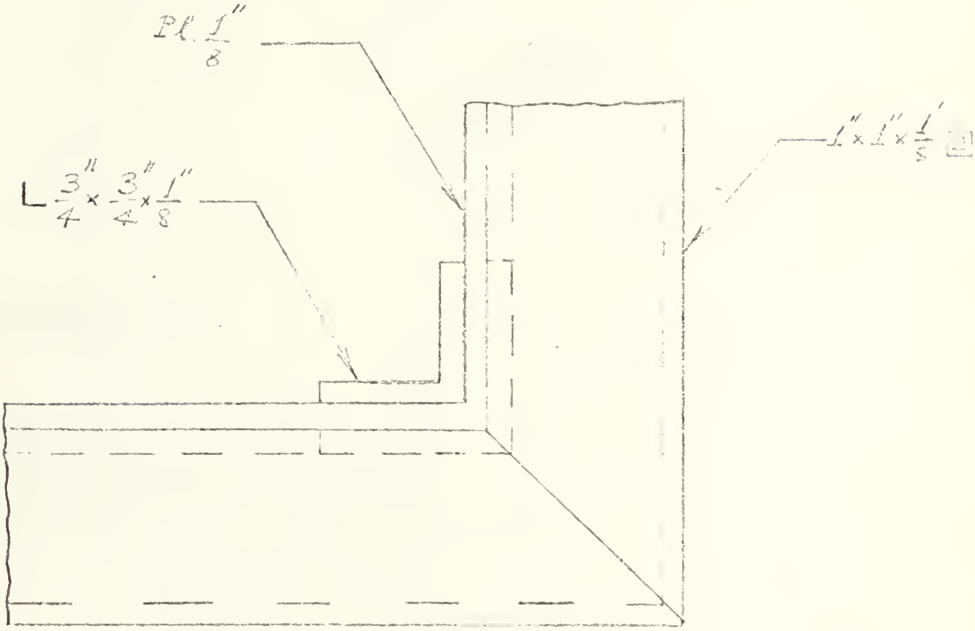
NOTE: Covered with
$\frac{1''}{8}$ diamond
aluminum plate



SCALE: 1" = 1'

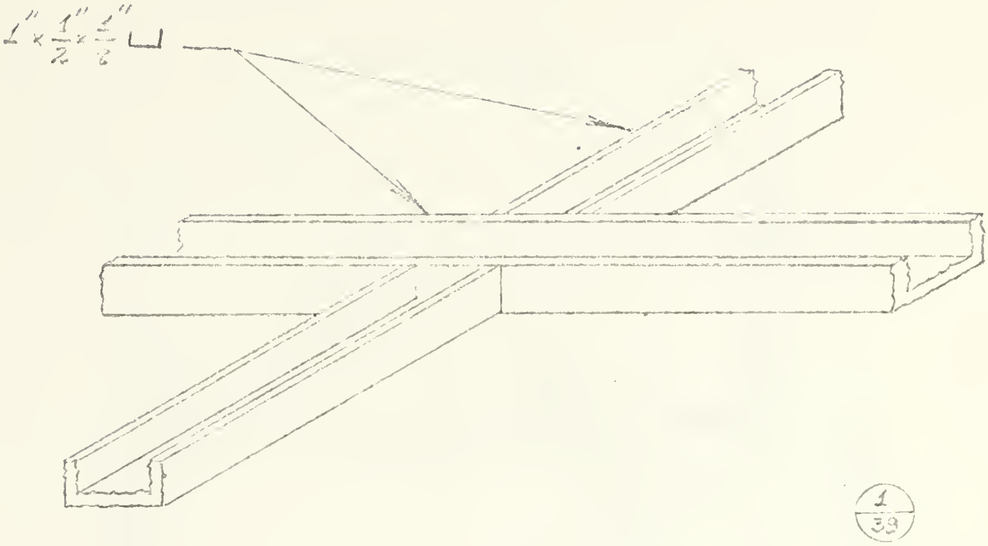
Fig. 37 Isometric View of Bucket Base

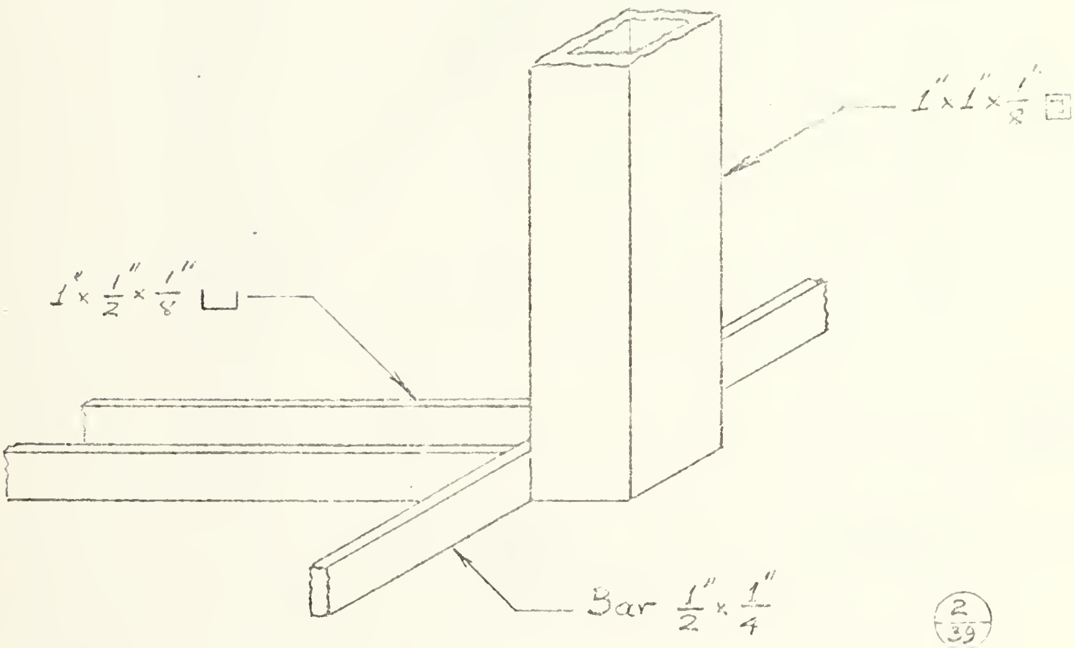




SCALE: 1" = 1'

Fig. 38 Details of Upper Corner of Bucket





SCALE: 1" = 2"

Fig. 39 Details of Bucket Base

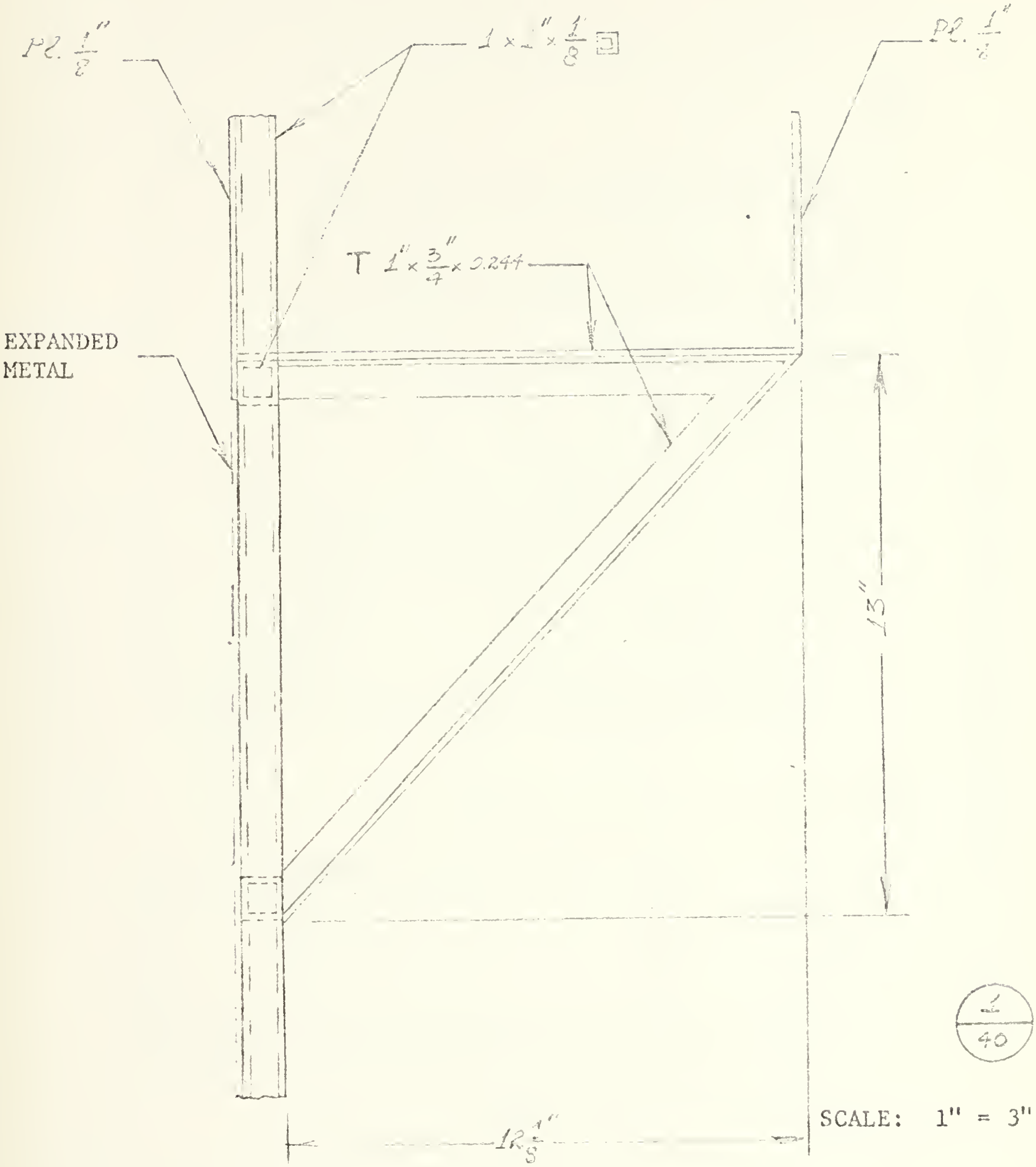
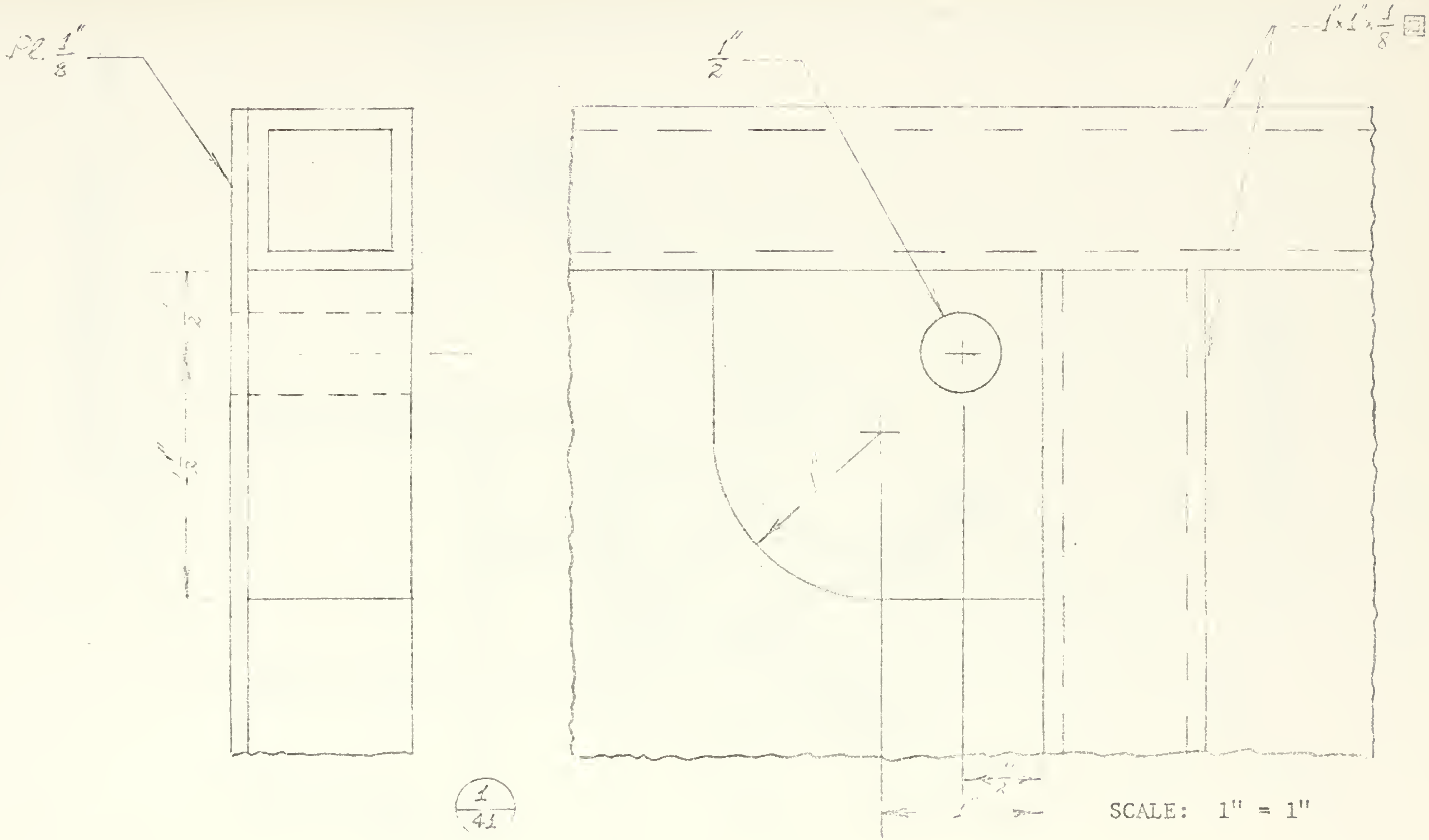


Fig. 40 Details of Storage Support



SCALE: 1" = 1"

Fig. 41 Magnetic Harness Pivoting Point on Bucket

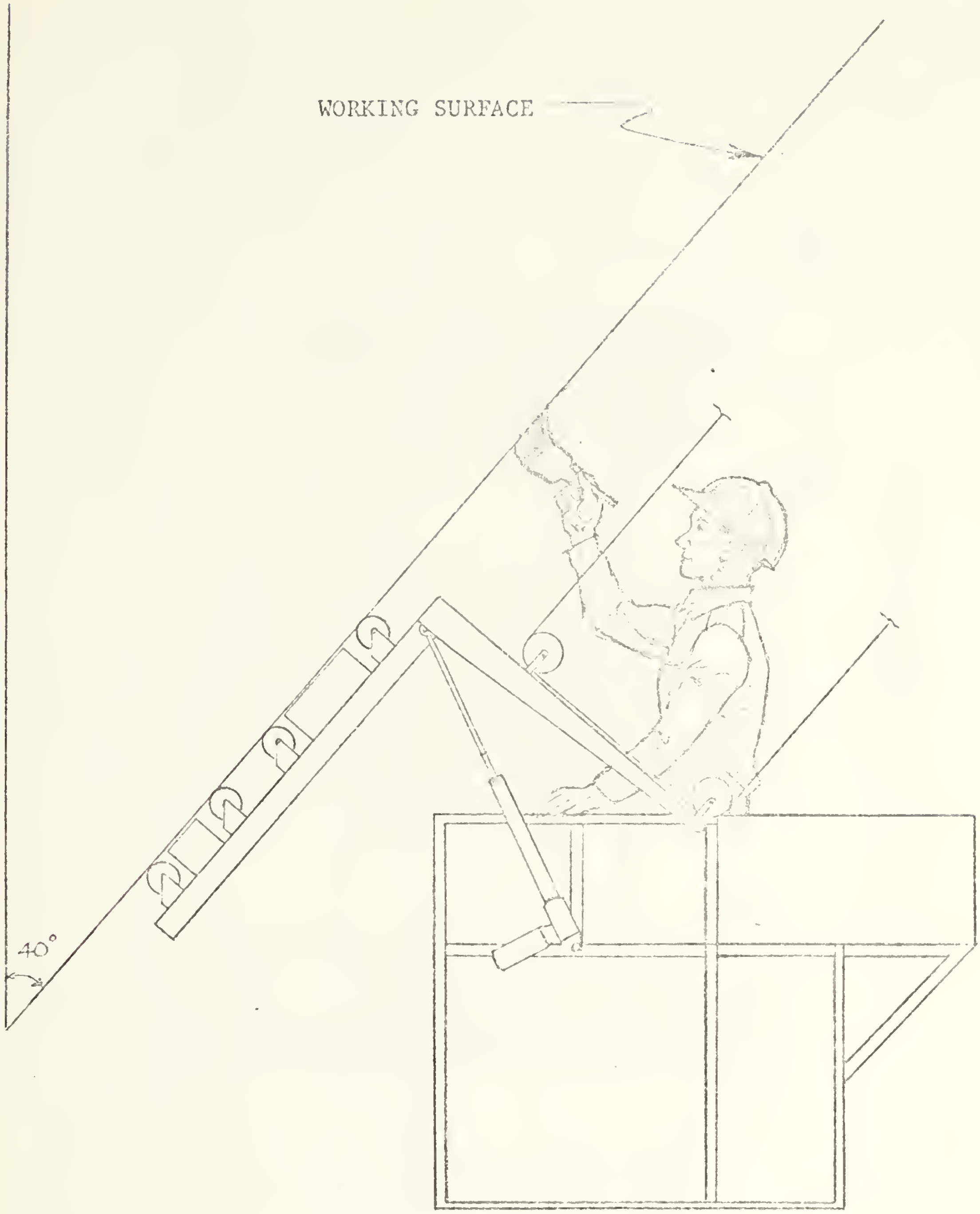


Fig. 42 Bucket and Magnetic Harness at Maximum Working Position

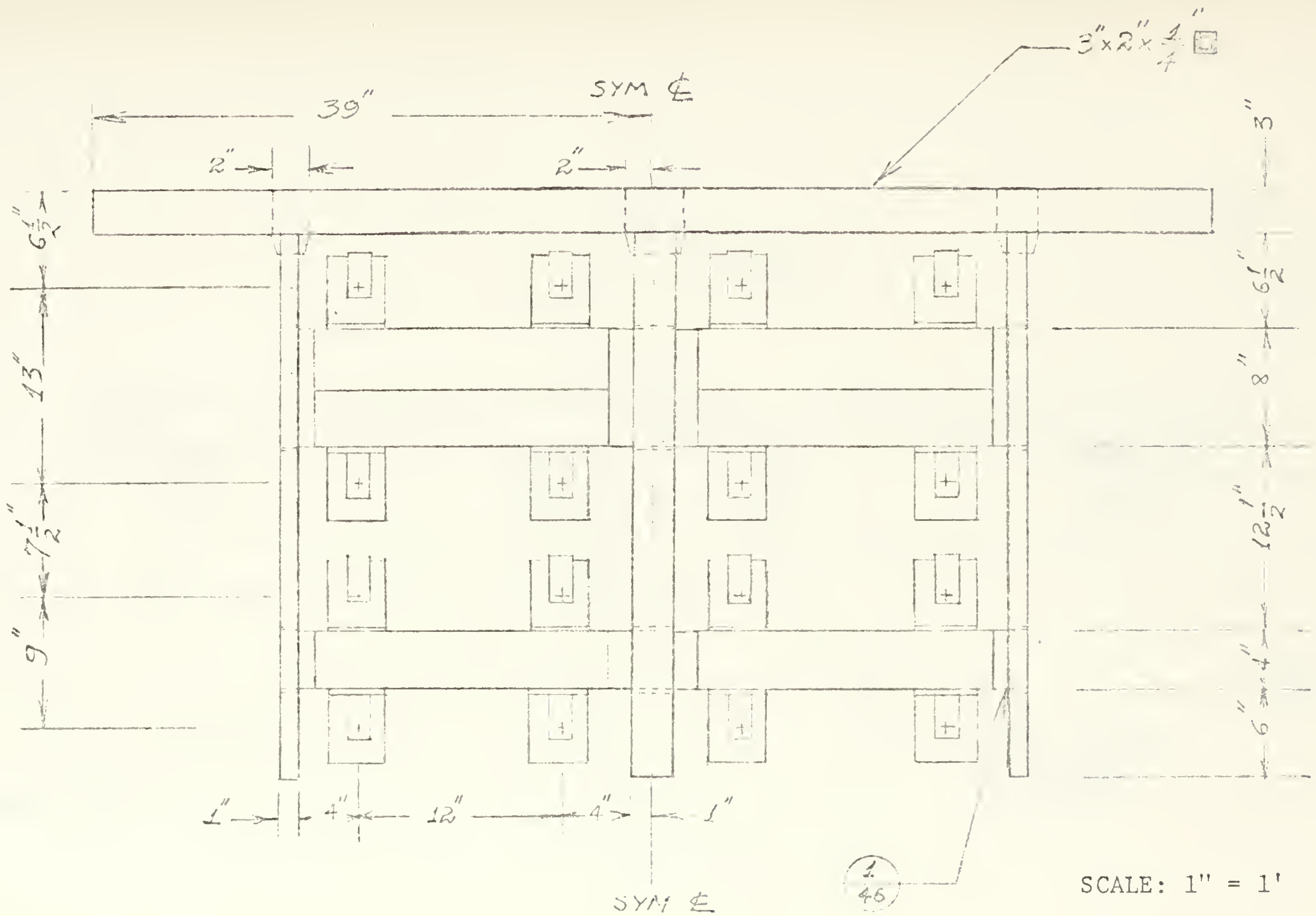
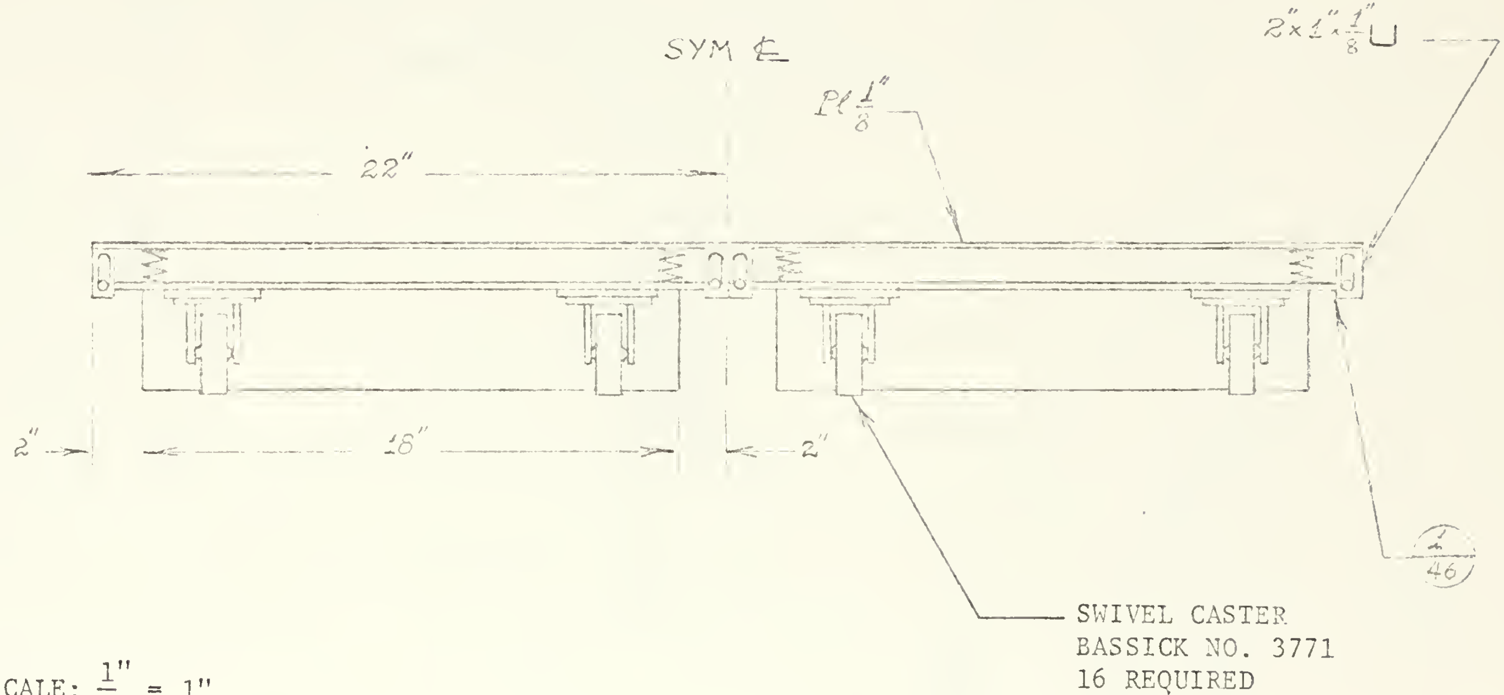


Fig. 43 Front View of Magnetic Harness



SCALE: $\frac{1''}{8} = 1''$

Fig. 44 Top View of Portion of Magnetic Harness Showing Suspension System

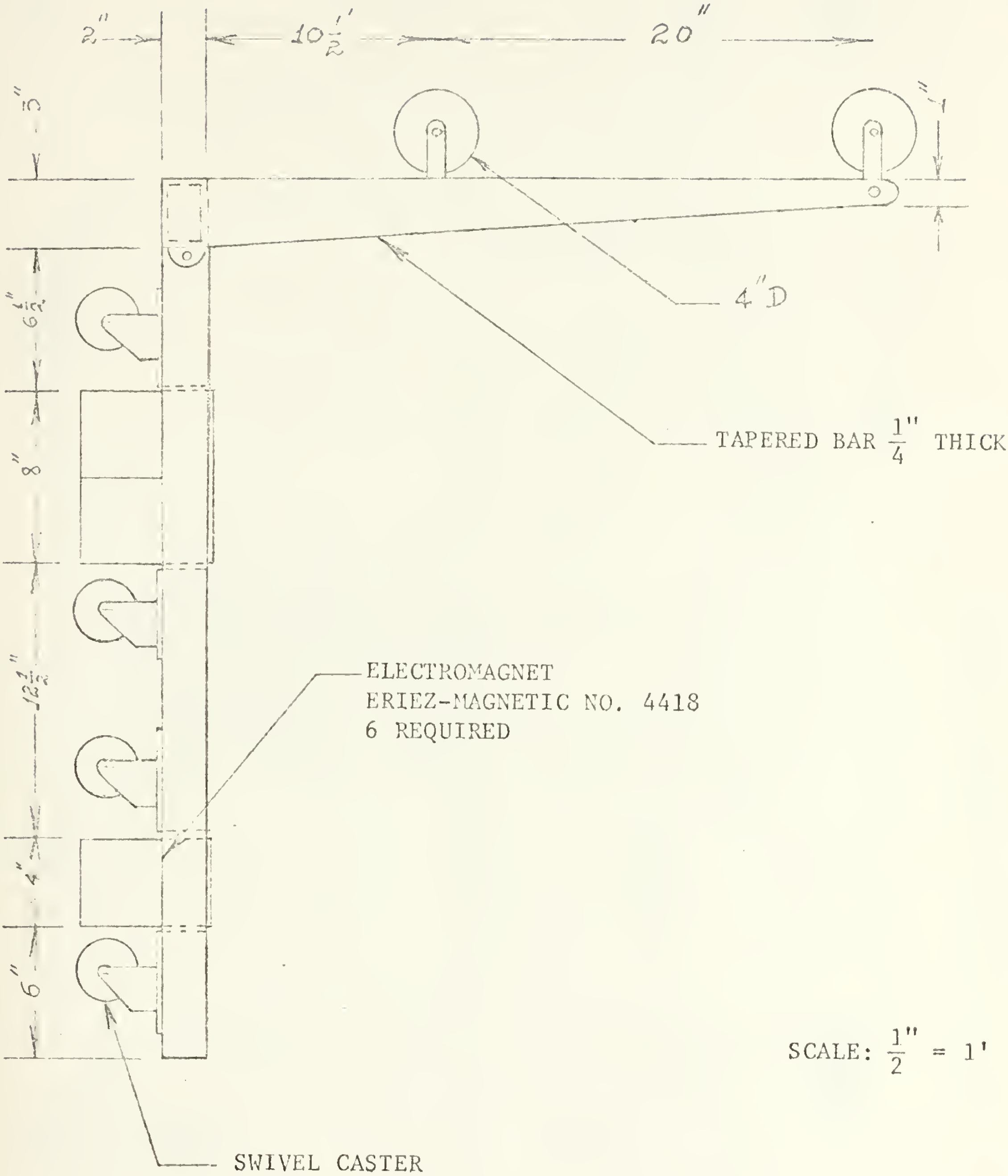
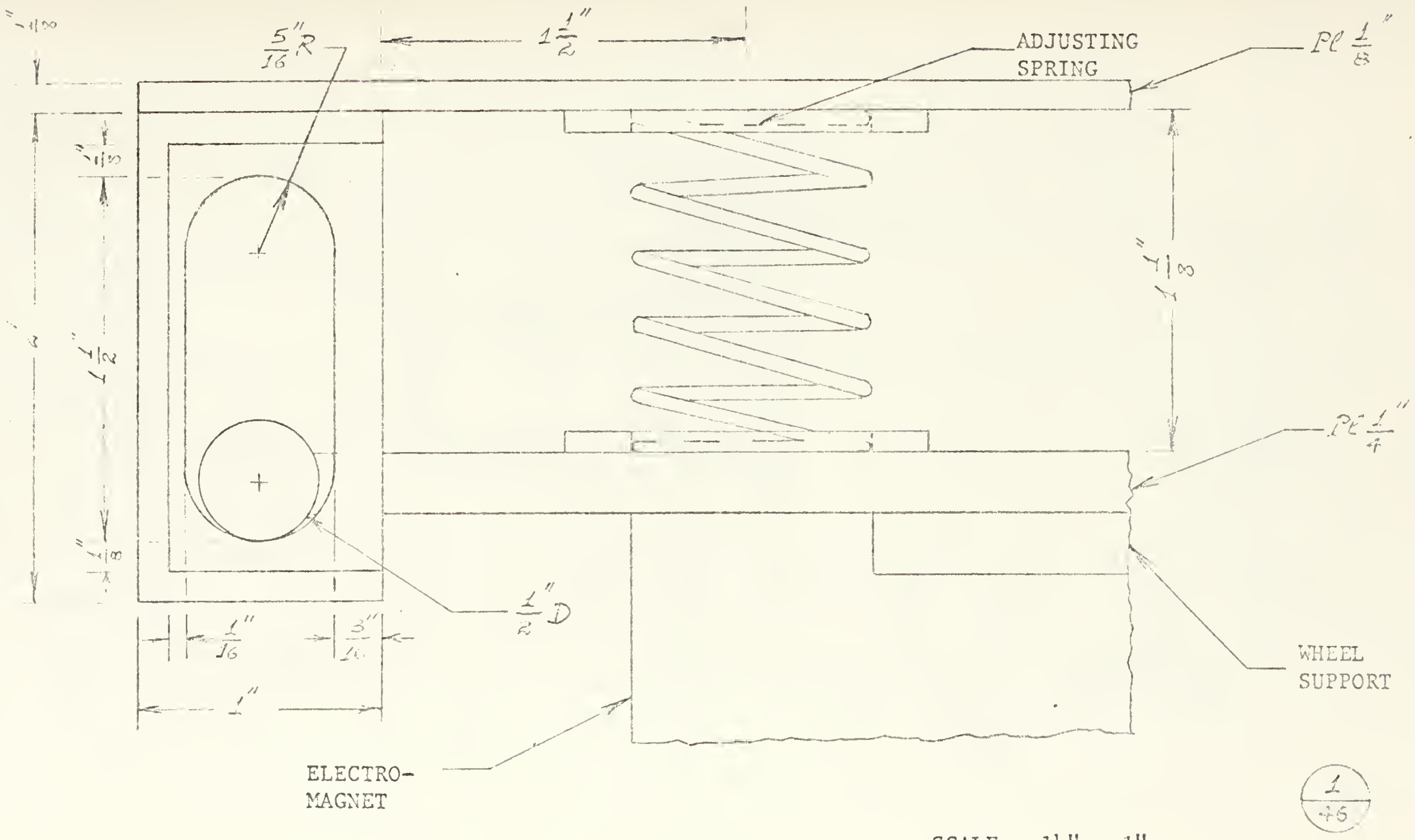
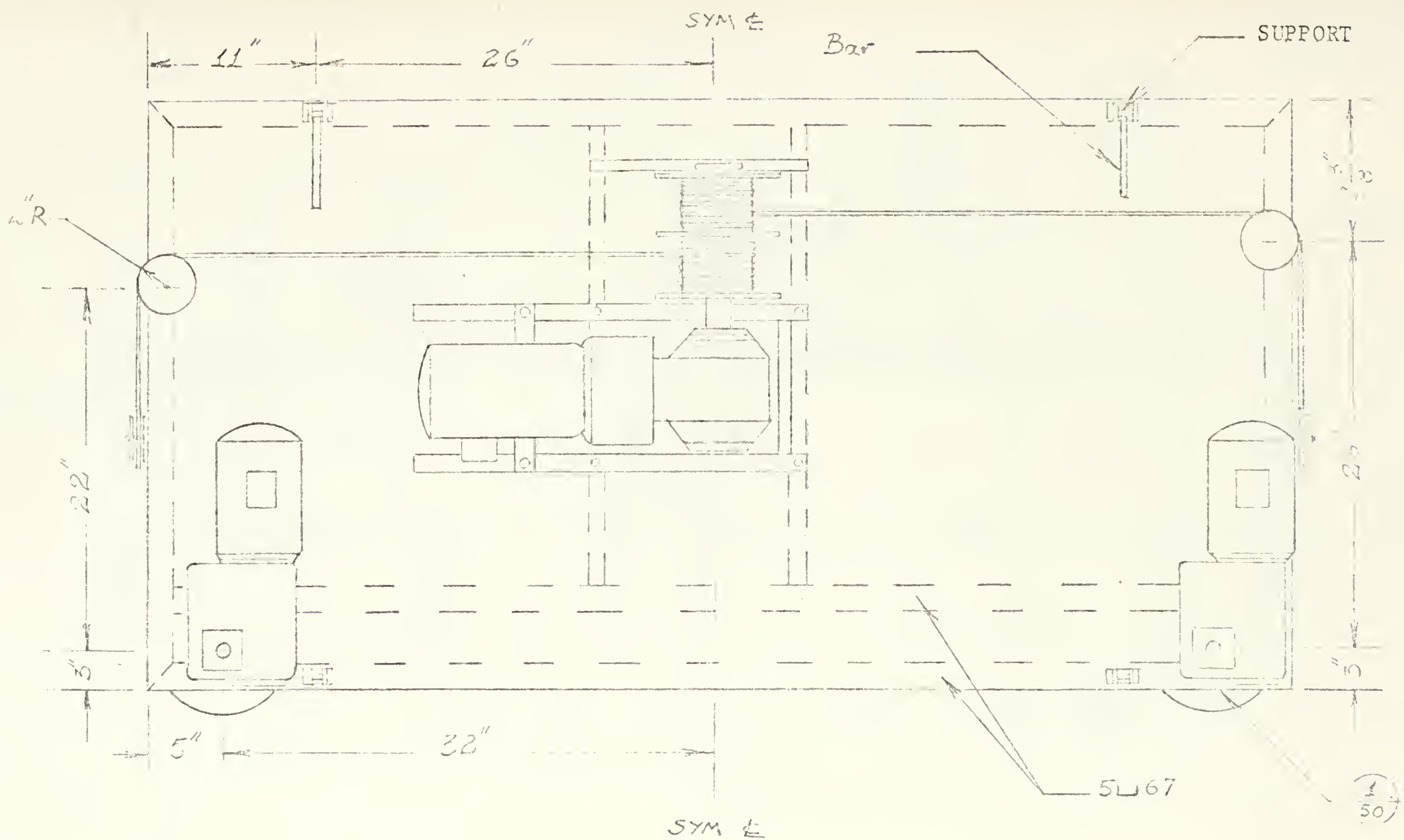


Fig. 45 Side View of Magnetic Harness



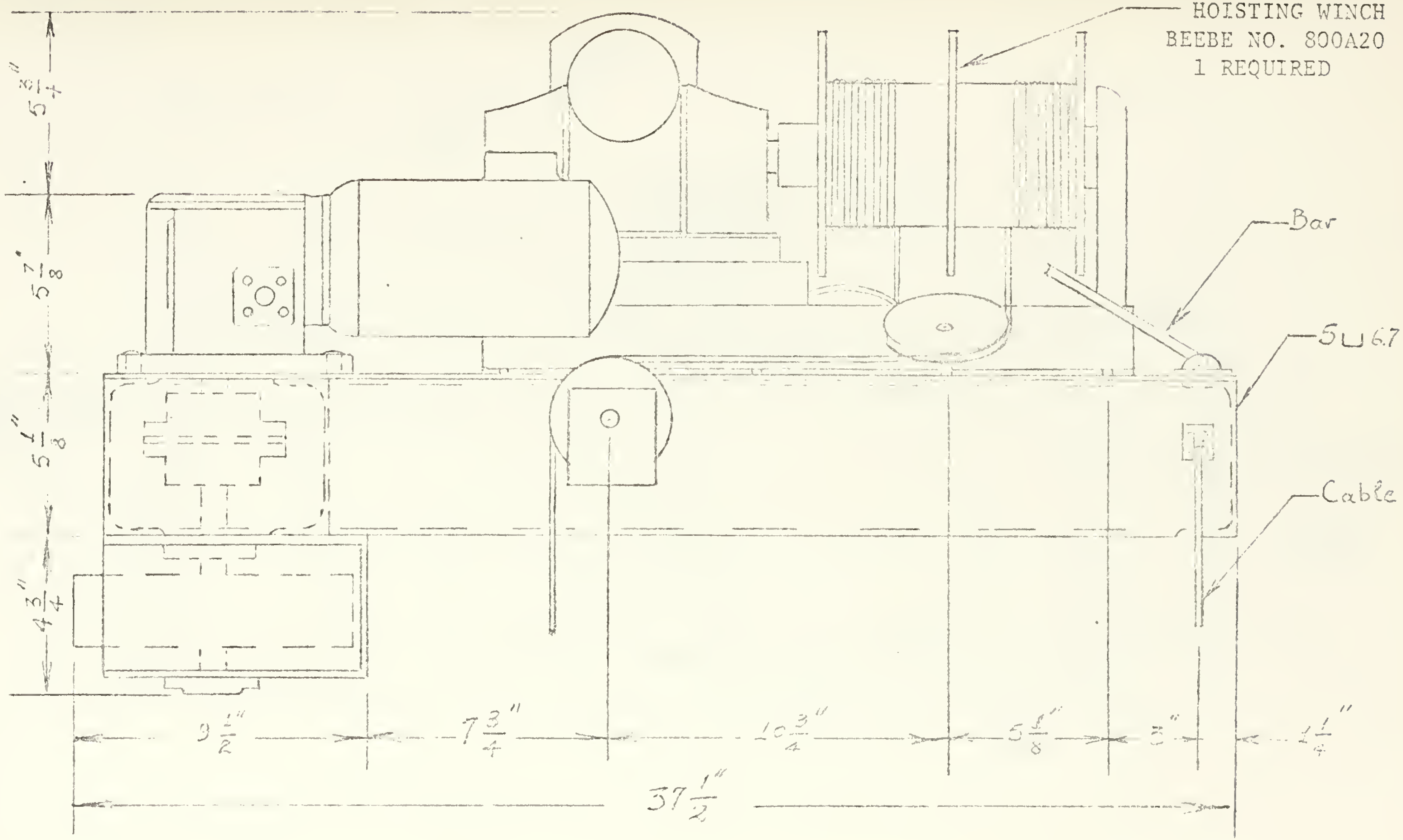
SCALE: 1½" = 1"

Fig. 46 Details of Magnet Support



SCALE: $\frac{3''}{32} = 1''$

Fig. 47 Top View of Moving Platform



SCALE: $\frac{3''}{16} = 1''$

Fig. 48 Side View of Moving Platform

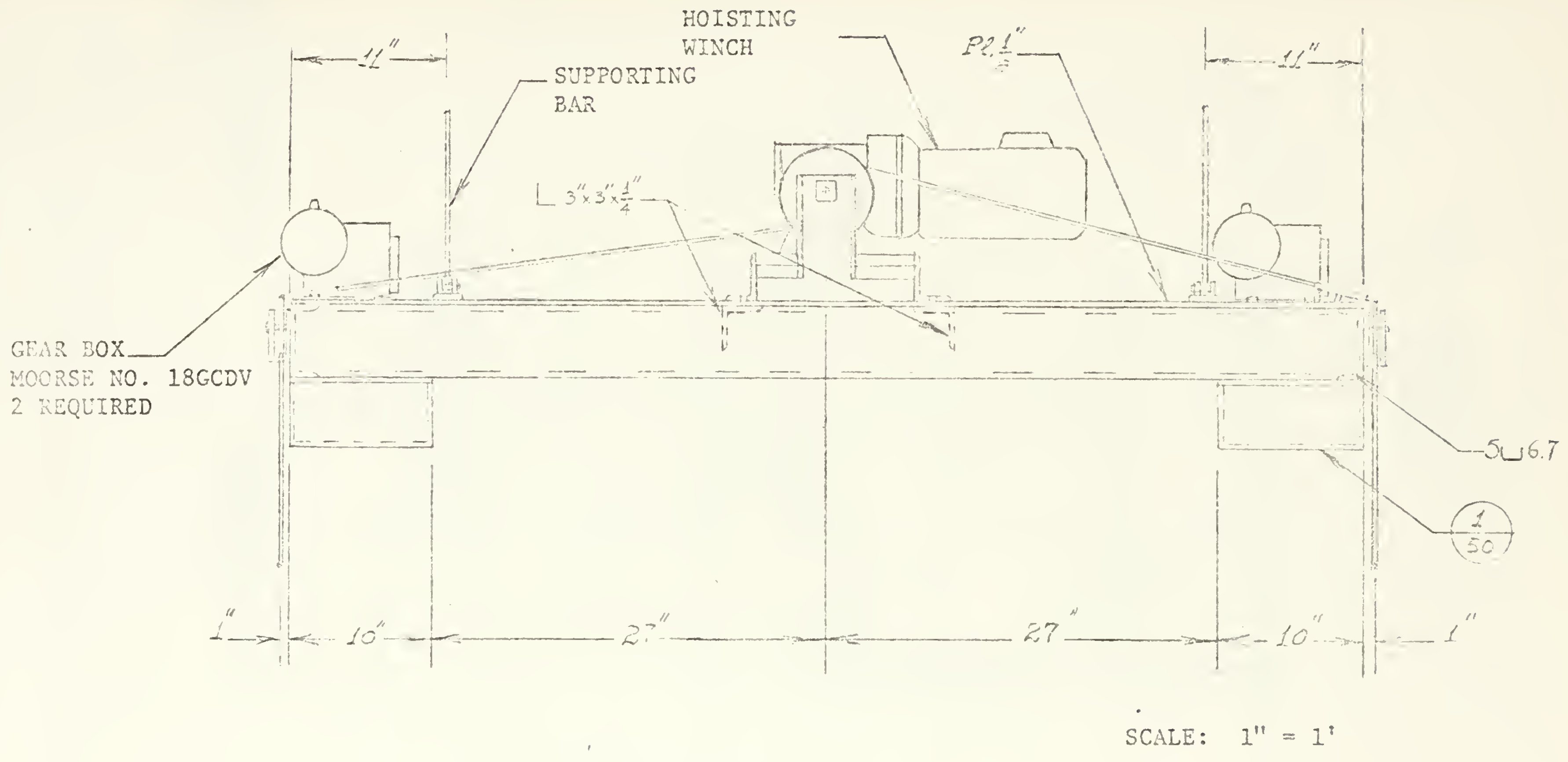


Fig. 49 Back View of Moving Platform

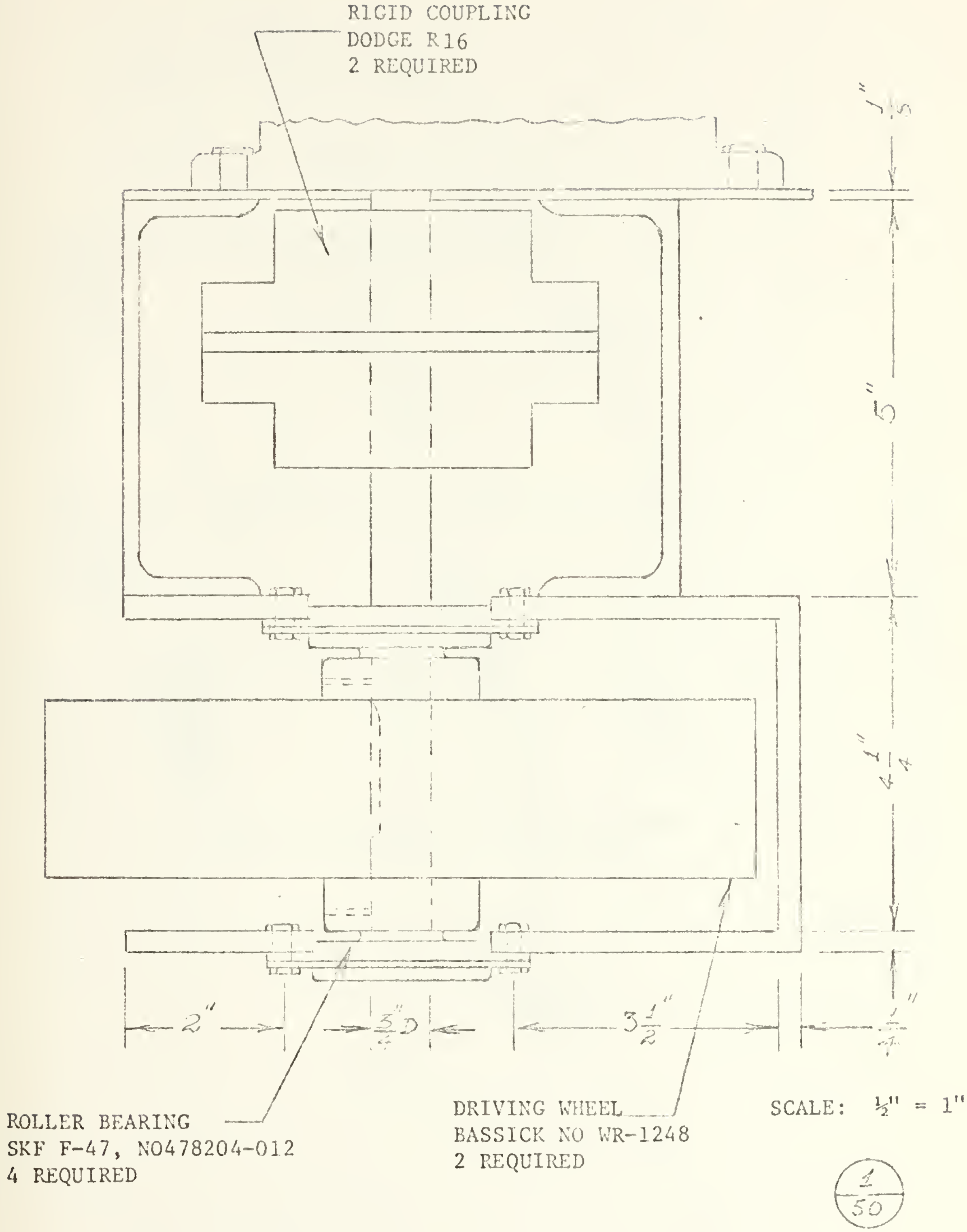
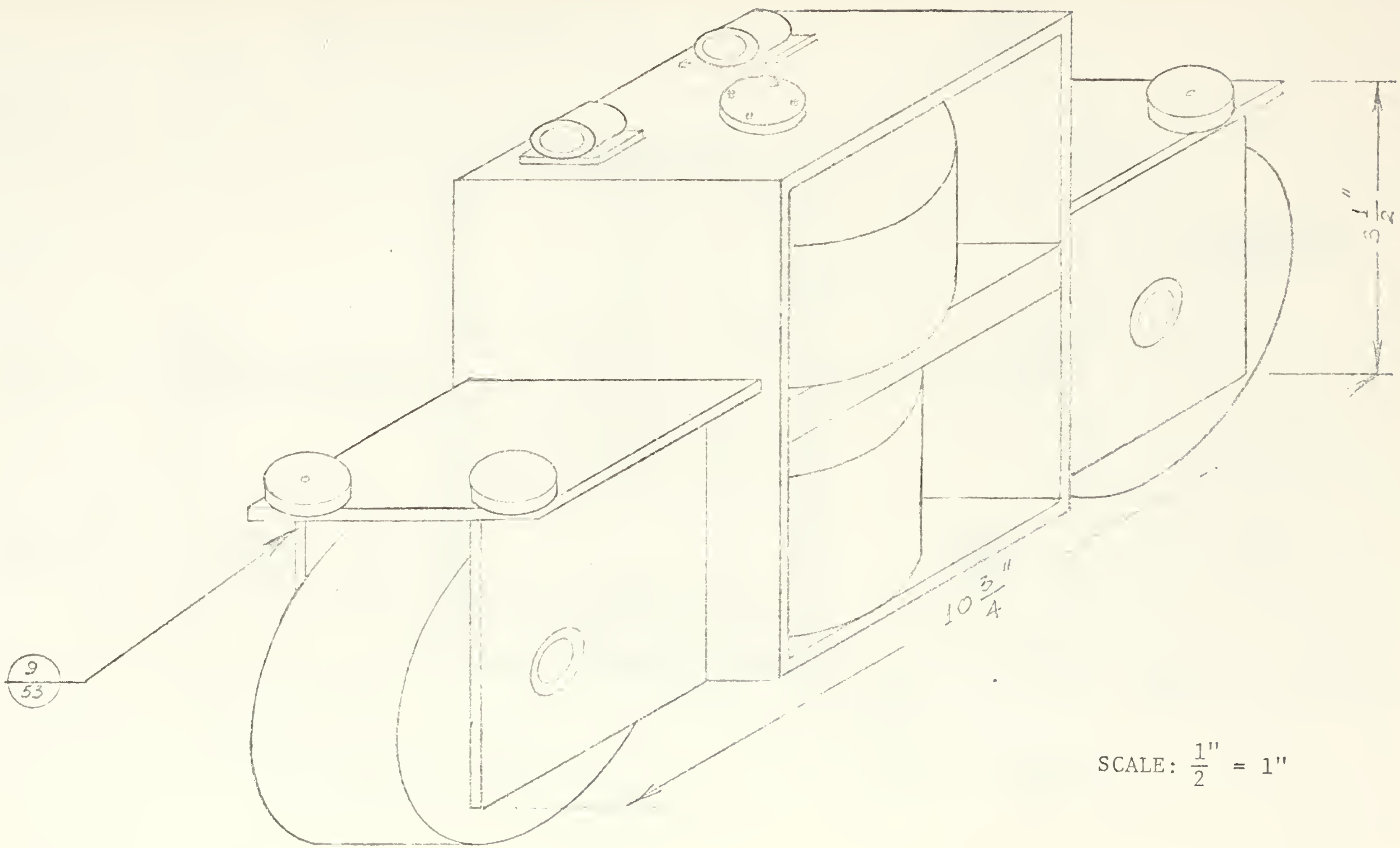


Fig. 50 Details of Driving Wheel Installation



SCALE: $\frac{1''}{2} = 1''$

Fig. 51 Three Dimensional View of Moving Support

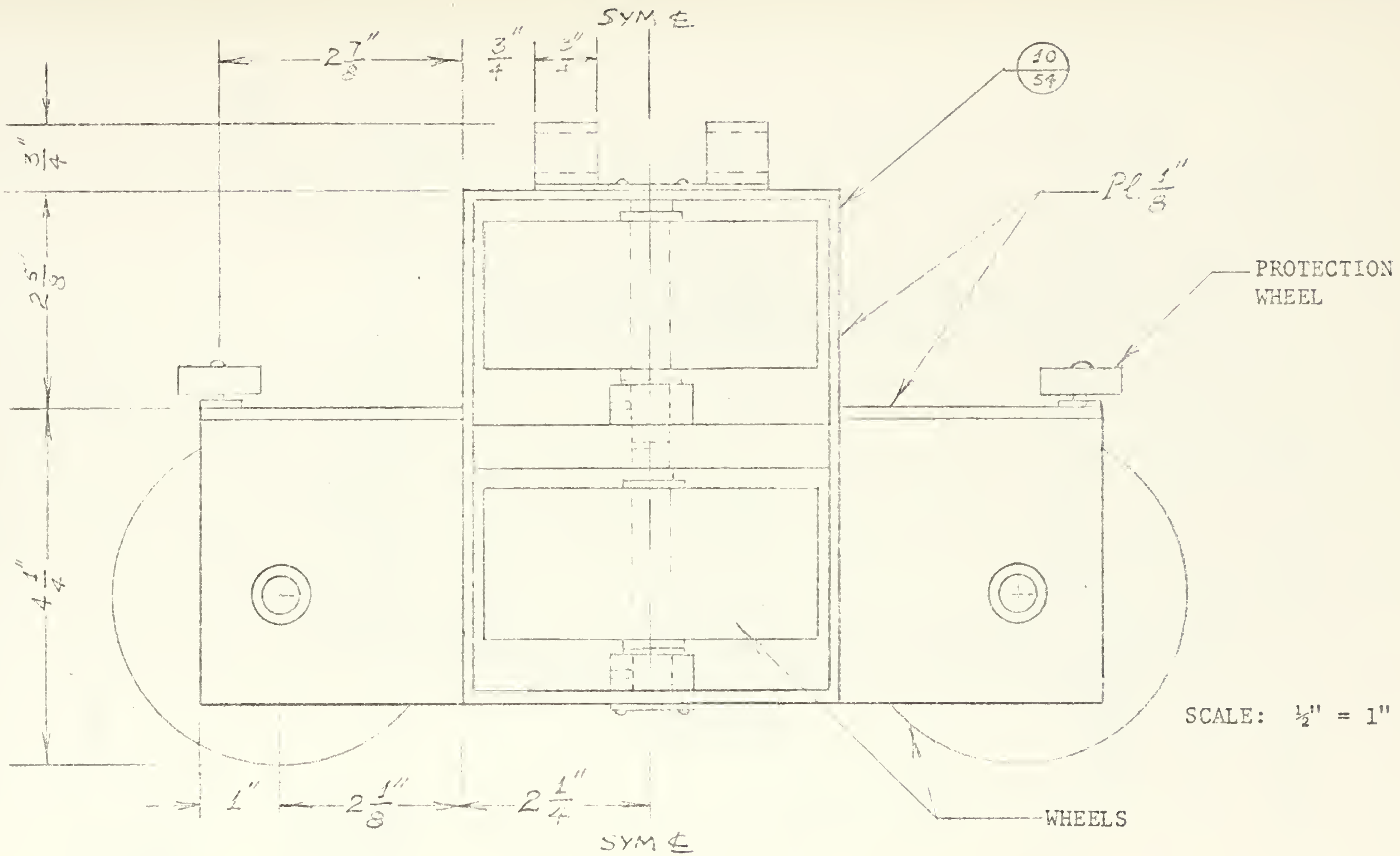


Fig. 52 Front View of Moving Support

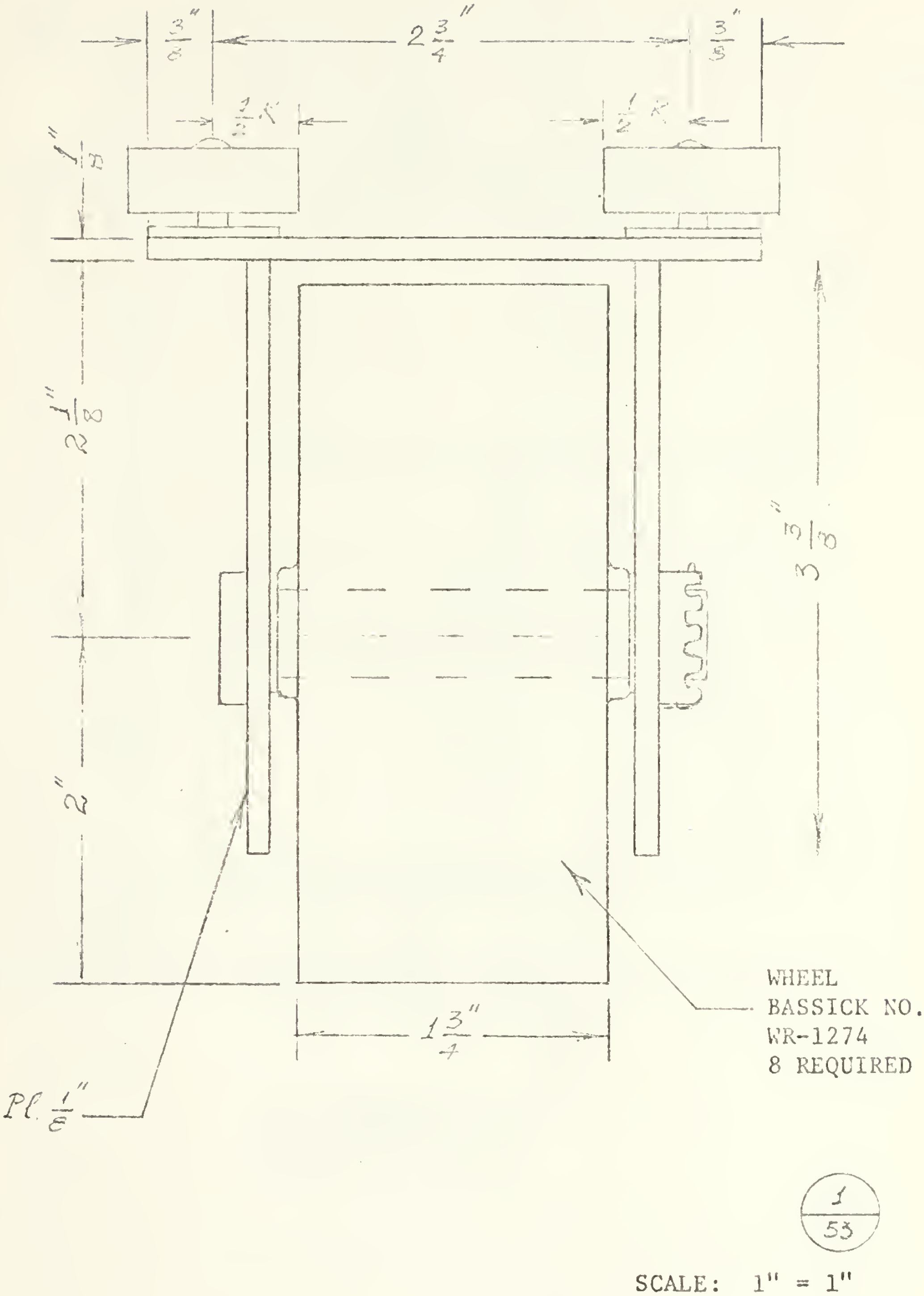
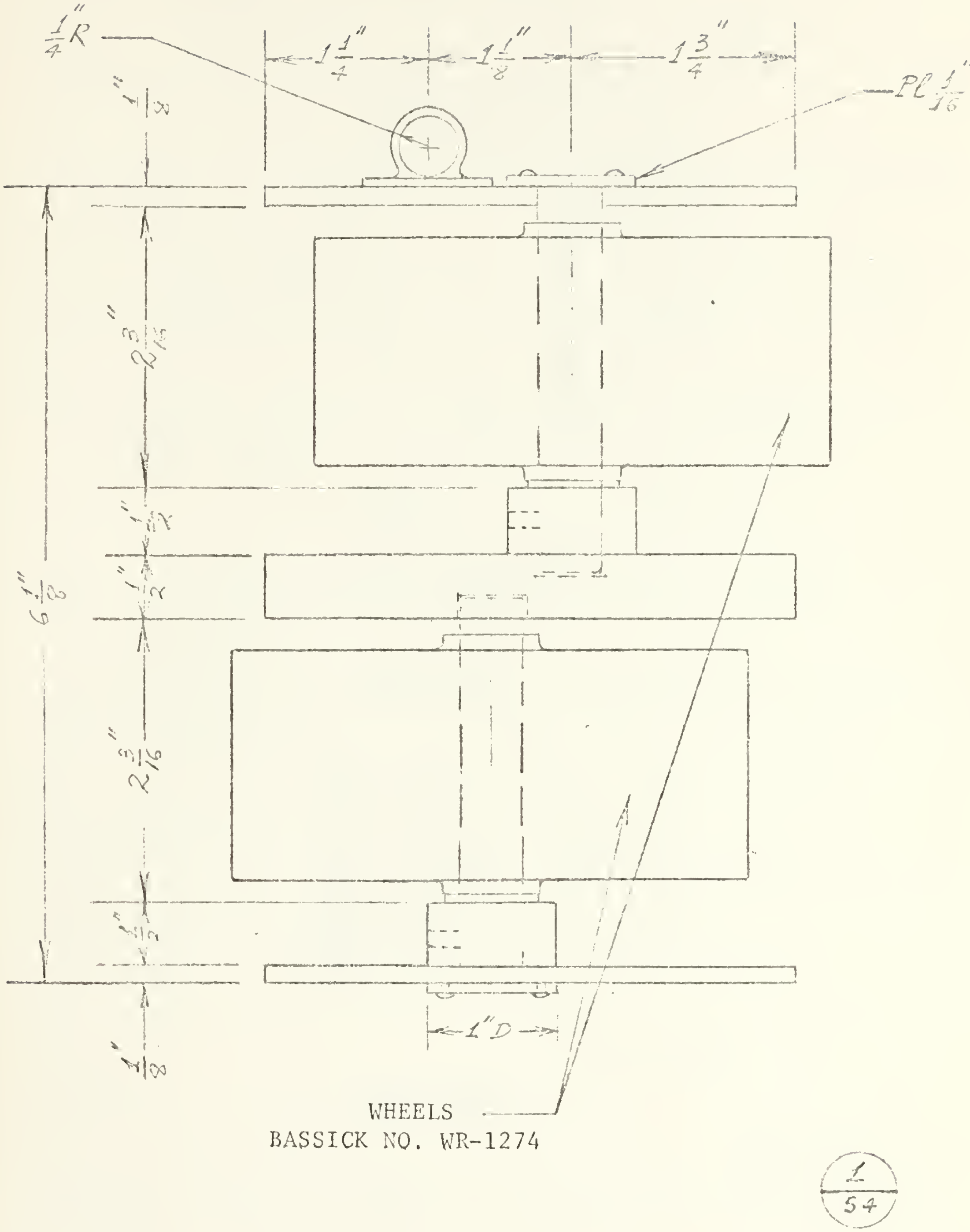
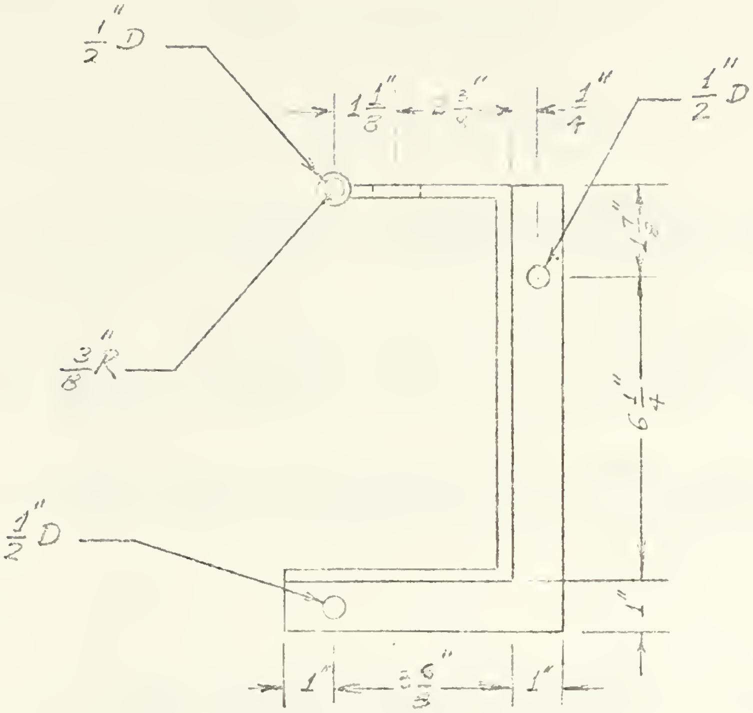


Fig. 53 Details of Vertical Wheel of Moving Support

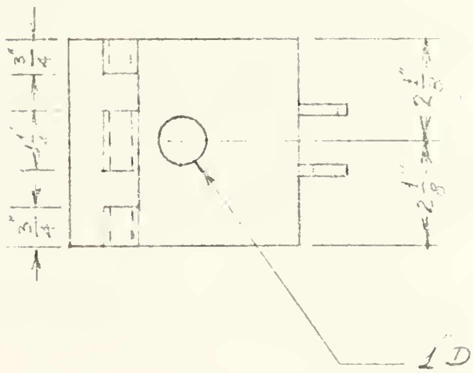


SCALE: $\frac{3}{4}$ " = 1'

Fig. 54 Section of Horizontal Wheels of Moving Support



SIDE VIEW



TOP VIEW

SCALE: $\frac{1''}{4} = 1''$

Fig. 55 Moving Support Hanger

LIST OF REFERENCES

1. Rossnagel, W. E., Handbook of Rigging, 3rd. ed., p. 207, McGraw-Hill, 1964.

2. Chicago Bridge and Iron Company, Subject: Water Tank Maintenance Techniques; Request for Information on, 22 January 1973.

3. Philadelphia Naval Shipyard Letter Code 383.1, Subject: Mobile Manned Work Platforms; Request for Information on, 22 February 1973.

4. Naval Undersea Research and Development Center, Report NVC TP 278, Ocean Engineering, p. 90, January 1972.

5. Hunter's Point Naval Shipyards, Personal Visit, February 1973.

6. Puget Sound Naval Shipyard, Letter Code 385.2/5720/1 Subject: Mobile Manned Work Platforms; Request for Information on, 27 February 1973.

7. Long Beach Naval Shipyard, Letter Ser. 70-08Q, Subject: Mobile Manned Work Platforms; Request for Information on, 2 March 1973.

8. Indiana General Co. Inc., Magnet Division, Short Cut for Holding Magnet Design, Form 371.

9. Sundstrand Machine Tool Co., Subject: Electromagnets; Influence of Underwater Operation; Request for Information on, 29 January 1973.

10. The Aluminum Association, Aluminum Standards and Data, 3rd. ed., January 1972.

11. Woodson, T. T., Introduction to Engineering Design, Appendix B, McGraw-Hill, 1966.

12. Eriez Magnetics, Letter, Subject: Performance of Magnet Models 44 and 66; Request for Information on, 17 May 1973.

13. Bassick Division, Stewart-Warner Corp., Catalog No. 128, 1973.

14. Mark's Standard Handbook for Mechanical Engineers, p. 3-35, 7th ed., McGraw-Hill, 1967.

15. Morse, Power Transmission Products, Catalog SP.71, p. E-191 and p. D-6, 7.

16. Dodge, Engineering Catalog, D70, p. 70-50.

17. SKF, Product Catalog, No. 450A.

18. Beebe Winches, Catalog No. SC-21, August 1971.

19. The Aluminum Association, Specifications for Aluminum Structures, Section 1, 2nd ed., November 1971.

20. Roark, J. R., Formulas for Stress and Strain, McGraw-Hill, 1965.

21. Popov, E. P., Introduction to Mechanics of Solids, p. 47, Prentice-Hall, Inc., 1968.

INITIAL DISTRIBUTION LIST

		No. Copies
1.	Library, Code 0212 Naval Postgraduate School Monterey, California 93940	2
2.	Dr. E. A. McKinnon, Code 59Mz Department of Mechanical Engineering Naval Postgraduate School Monterey, California 93940	12
3.	Hellenic Naval Command Stratopedon Papagou, Holargos Athens, Greece	1
4.	Lt. Constantine Z. Pagonaris 133-135 Papagou Av. Zografou - Athens - Greece	1
5.	Defense Documentation Center Cameron Station Alexandria, Virginia 22314	2
6.	Dr. Robert Nunn, Code 59Nn Chairman, Department of Mechanical Engineering Naval Postgraduate School Monterey, California 93940	1
7.	Prof. R. W. Prowell, Code 59Pr Department of Mechanical Engineering Naval Postgraduate School Monterey, California 93940	1

SECURITY CLASSIFICATION OF THIS PAGE (When Data Entered)

REPORT DOCUMENTATION PAGE		READ INSTRUCTIONS BEFORE COMPLETING FORM
1. REPORT NUMBER	2. GOVT ACCESSION NO.	3. RECIPIENT'S CATALOG NUMBER
4. TITLE (and Subtitle) Preliminary Design of a Self-Propelled Manned Work Platform		5. TYPE OF REPORT & PERIOD COVERED Master's Thesis; September, 1973 6. PERFORMING ORG. REPORT NUMBER
7. AUTHOR(s) Constantine Pagonaris		8. CONTRACT OR GRANT NUMBER(s)
9. PERFORMING ORGANIZATION NAME AND ADDRESS Naval Postgraduate School Monterey, California 93940		10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS
11. CONTROLLING OFFICE NAME AND ADDRESS Naval Postgraduate School Monterey, California 93940		12. REPORT DATE September 1973 13. NUMBER OF PAGES 110
14. MONITORING AGENCY NAME & ADDRESS(if different from Controlling Office) Naval Postgraduate School Monterey, California 93940		15. SECURITY CLASS. (of this report) Unclassified 15a. DECLASSIFICATION/DOWNGRADING SCHEDULE

16. DISTRIBUTION STATEMENT (of this Report)

Approved for public release; distribution unlimited.

17. DISTRIBUTION STATEMENT (of the abstract entered in Block 20, if different from Report)

18. SUPPLEMENTARY NOTES

19. KEY WORDS (Continue on reverse side if necessary and identify by block number)

Maintenance Platform

20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

The maintenance operations of most large steel structures are currently being performed from permanent or movable platforms. The trend toward minimizing the cost resulted in a demand for a self-propelled movable platform that is controlled by the maintenance personnel on the platform. In this project a self-propelled work platform was created and designed which is capable of following the contour of a steel structure. The study

DD FORM 1 JAN 73 1473 EDITION OF 1 NOV 65 IS OBSOLETE
(Page 1) S/N 0102-014-6601 |

SECURITY CLASSIFICATION OF THIS PAGE (When Data Entered)

20. was oriented toward naval applications where, in general, the ship hulls present curved surfaces which are often inclined with respect to the vertical. The proposed system can also be used or easily modified for non-naval applications.

DD Form 1473 (BACK)
1 Jan 73
S/N 0102-014-6601

a

Thesis 146183
P1165 Pagonaris
c.1 Preliminary design of a
self-propelled manned
work platform.